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BULLETIN
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Graphical method

of determining the frictional and guiding forces on railway vehicles running through curves,

by Reichsbahnoberrat Dr.-Ing. TROITZSCH, RBD Wuppertal.

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* * *

A) Preface.

The problem of ascertaining the forces exerted on the vehicle when running through a curve has been closely investigated, especially by ÜBELACKER, HEUMANN, JÄHN and BÄSELER, whose investigations have thrown light on the forces which occur when running through a curve, a matter which had long been in doubt. Excepting HEUMANN, they all solve the problem purely by calculation. This method is in most cases very arduous, lengthy and not easy to check.

We have, therefore, endeavoured to give below a graphical method, which is generally clearer and leads more easily to results, and in addition, the laws in question are more clearly recognised. This graphical method is based on BÄSELER's papers : « Die Spurkranzreibung » (The flange friction) (1) and « Das Geheimnis der freien Lenkachsen » (The mystery of the free-radial axles) (2). Our method can be used for all cases occurring in practice.

If a suitable scale is used, the same accuracy can be obtained as by calculation.

Making use of these two papers, the analysis is limited to two-point contact of the striking wheels.

* * *

B) Notations and simplified assumptions.

The notations used in table 1 will be used in this paper; they are the same as those in BÄSELER's papers.

TABLE 1.

List of notations used throughout this article.

I	= first axle (leading pair of wheels).
II	= second axle from the front.
III	= third axle from the front.
1, 2, 3	= outside wheel of axle I, II, III.
1′, 2′, 3′	= inside wheel of axle I, II, III. These numbers are also used as indices. The addition of the apostrophe means the inside of the curve.
Q	= average wheel pressure (weight of the complete vehicle divided by number of wheels).
$Q_{1,2,3}$	= Actual wheel pressure occurring at the outside wheel of axle I, II, III.
N	= Pressure of the rail head at its upper curvature, nearly perpendicular, against the wheel ($N_i = Q_i$, if no striking of the wheel occurs at point i).
$\mathfrak{R}$	= Normal pressure at point of contact $\mathfrak{A}$ of wheel and rail. $\mathfrak{R}$ is inclined to the horizontal by $30° = 90° - \beta$ for the profile assumed for rail and wheel flange.
P	= $\mathfrak{R}_y$ = directing force = horizontal component of $\mathfrak{R} = \mathfrak{R} \sin \beta$.
$\mathfrak{R}_z$	= Vertical component of $\mathfrak{R} = \mathfrak{R} \cos \beta$.
	= Wheel bearing point.
$H_{1,2...}$	= Resultant of the frictional forces occurring at contact point of the two wheels of an axle, which is parallel to the direction of the wheels.
$\mathfrak{A}$	= Flange pressure point of striking wheel [here assumed 8 mm. (5/16") below rail top].
$f\mathfrak{R}$	= Frictional force at the point $\mathfrak{A}$.
$\mathfrak{F}$	= Vertical component of $f\mathfrak{R}$.
$\mathfrak{S} \overset{\wedge}{+} \mathfrak{K}$	= Component of $f\mathfrak{R}$ in the horizontal plane.
α	= Striking angle of pair of wheels in the curve.
$H_{I.II,..}$	= Horizontal pressure in transverse direction of the axle-guard against the axle.
fN	= Approximately horizontal frictional force at the point of contact of wheel and rail head.
G	= Component of fN, vertical to the longitudinal centre line of the vehicle (Y component).

(1) *Organ*, 1927, No. 18.

(2) *Z. V. D. E. V.*, 1929, No. 14.

K = Component of fN in the direction of the longitudinal centre line of the vehicle (X component).

V = Speed of the vehicle, in km./h.

v = Speed of the vehicle, in m./sec.

$\mathfrak{v}$ = Velocity of the flange pressure-point against the side bearing point $\mathfrak{A}$ at the rail-head. $\mathfrak{v}$ is split up into the directions X (parallel to the longitudinal centre line of the vehicle), Y (parallel to the centre line of the wheel) and Z (perpendicular to the longitudinal centre line of the vehicle and that of the wheel at $\mathfrak{v}_x$, $\mathfrak{v}_y$ and $\mathfrak{v}_z$.

b = Lead of the point $\mathfrak{A}$ relatively to A, on the striking wheel.

$\pm \Delta R$ = Distance of the centre of friction from the longitudinal centre line of the vehicle (+ inwards, — outwards).

M = Centre of friction.

$\mathfrak{M}$ = Static moment; $\mathfrak{M}_{I,II,III,...}$ = static moment of the forces acting on the axles I, II, III.

p = Distance of the centre of friction M from the striking pair of wheels.

l = Wheelbase.

R = Radius of the track curve.

r = Radius of the wheel, at the tread circle.

s = Distance between the wheel contact points, assumed to be 1.50 m. (4' 11").

P_s = Outer lateral force.

Z = Tractive effort at the drawbar hook.

S = Braking effort.

R_a line = —.—.—.— = Line representing the resultant in the horizontal plane of all forces acting at point $\mathfrak{A}$.

R_n line = —..—..—..— = Auxilary line = a special geometrical locus in the force diagram.

The following scales are recommended:

For the location plan (layout) 1 : 10.

For the force diagram Q = 40 cm. (15 3/4"), or 1 cm. = $\frac{Q}{40}$.

At first the conical form of the wheel tyre is not considered, i. e. the problem is treated as if the wheel tyres were exactly cylindrical. The influence of the conical shape on the occurring forces is not of any decisive importance with the exception of free radial axles and will be considered when dealing with them.

The following considerations are based on the assumption of the old S 49 rail section and the V. M. E. V. (Union of Central European Railways) type of flange, the standard type now used by the German State Railways.

All the occurring forces are expressed in terms of the average wheel pressure Q (see Table 1).

The rolling resistance of the wheels is neglected with respect to the forces occurring elsewhere.

In all other respects, the assumptions made in the above-mentioned papers of Bäseler shall also apply here. The bearings are considered as frictionless, the rotation around the centre 0 of curvature takes place at a constant speed, the tread circles of all wheels considered have the same radius, and the conical shape of the wheel tyres is therefore neglected; the distance between the tread circles of the two wheels of a pair is assumed as 1.50 m.; the very small striking angle α is a negligible quantity when compared with the angles formed by the frictional forces with the longitudinal axis of the vehicle, and can be disregarded when draw-

ing the directions of the forces; the position of the flange pressure point $\mathfrak{A}$ is assumed leading only in the plane of the vehicle in the forward direction without taking any lateral displacement into account. This flange pressure point, for new rails and wheels, is taken as 8 mm. (5/16") below the rail top, assuming cylindrical wheel tyres, and as 9.1 mm. (3/8") below the rail top when considering cone-shaped wheel tyres.

Vehicles with rigid axles, with and without lateral play of the axles, are considered hereafter.

For the sake of brevity, kinetic problems and details, especially the motion conditions at the flange, are only mentioned as far as appears absolutely necessary for comprehension. Here again reference must be made to well-known papers, especially that on « Flange friction » by BÄSELER, for details. The equations of equilibrium are also given therein for the pairs of wheels striking under two-point contact, which have been omitted here.

* * *

C) General remarks about the graphical method.

The wheels of the vehicle running under steady working conditions, through a curve, do not only roll, but slide at the same time. The slip of each pair of wheels takes place round its « centre of friction » M (1). For vehicles with rigid axles this centre is on the perpendicular from the centre of curvature to the longitudinal axis of the vehicle. To illustrate this, we shall consider how the vehicle shown in figure 1, as a general example, proceeds with the left front wheel guided by the outside rail with either the trailing pair of wheels also guided by the inside rail, or in a state of inertia with free trailing wheels, describing circles concentric to the rail curve. If the vehicle proceeds by an infinitely small distance, point 1 moves to point 1_b, point 1' to point $1'_b$, point 2 to point 2 , and point 2' to point $2'_b$. The centre of rotation of the whole vehicle is, of course, the centre 0 of the rail curve. Each distance $1\text{-}1_b$, $1'\text{-}1'_b$, $2\text{-}2_b$, and $2'\text{-}2'_b$, is split up into a frictionless pure rolling motion *dx*, and a residuary motion only occurring at the cost of the frictional effort $1\text{-}1_c$, $1'\text{-}1'_c$, $2\text{-}2_c$, $2'\text{-}2'_c$. Having assumed equal tread circles and rigidly mounted axles which can only be shifted parallel to themselves, the distance of the pure rolling, *dx*, is parallel to the centre line of the vehicle. From the similarity of the triangles $2\text{-}2_b\text{-}2_c$ and $2\text{-}0\text{-}M_{II}$, $2'\text{-}2'_b\text{-}2'_c$ and $2'\text{-}0\text{-}M_{II}$, $1'\text{-}1'_b\text{-}1'_a$ and $1'\text{-}0\text{-}M'_{Ia}$, etc., it follows that :

$$\frac{\overline{2—2_c}}{\overline{2—M_{II}}} = \frac{\overline{2—2_b}}{\overline{2—0}} = \frac{\overline{2—0}.d\varphi}{\overline{2—0}} = d\varphi$$

$$\overline{2—2_c} = \overline{2—M_{II}} \cdot d\varphi; \text{ similarly } \overline{2'—2'_c} = \overline{2'—M'_{II}} \cdot d\varphi; \ \overline{1'—1'_a} = \overline{1'—M'_{Ia}} \cdot d\varphi.$$

(1) ÜBELACKER : « Untersuchungen über die Bewegung von Lok. mit Drehgestellen in Bahnkrümmungen » (Research on the motion of bogie locomotives in track curves), *Organ*, 1903, Supplement.

Furthermore $\overline{0-M_{II}} = \frac{dx \cdot \overline{2-0}}{\overline{2-2_b}} = \frac{dx}{d\varphi}$, similarly $\overline{0-M'_{II}} = \overline{0-M'_{I}} = \frac{dx}{d\varphi}$;

$$\overline{0-M'_{Ia}} = \frac{dx_a}{d\varphi}$$

The first group of equations indicates that for a given constant running speed all rigid axles for which cylindrical flanges have been assumed will rotate round a track curve (after deducting the pure rolling which is of equal magnitude for both wheels of any one pair) with the same angle $d\varphi$, round their centre of friction M, as the complete vehicle will rotate round O, and that this centre of friction M is on the parallel to the axle through O.

The second group of equations shows that all pairs of wheels with pure rolling runs of equal magnitude, have a common centre of friction M (M_{II}, M'_{II}, M'_{I}, M_{I}) which also lies on the parallel to the axles through O. This is the case with coupled pairs of wheels, which are subjected to an equal rolling speed by the coupling rods.

If the pure rolling of one axle is larger or smaller than that of another (at axle II = dx, at axle I = dx_a), then the two pairs of wheels have two different centres of friction M, both of which, however, will be located on the parallel through O, ($M_{II} = M'_{II}$ and $M_{Ia} = M'_{Ia}$).

Under the assumptions made for wheels with side pressure roller guiding for which only a horizontally directed pressure occurs, exactly perpendicular to the rail at the side of the guiding rail, and on the assumption of equal pressures for all wheels, M is located on the point of intersection of the axle parallel with the longitudinal axis of the vehicle.

The first object to be attained is still to ascertain the position of the axle parallel, i. e. the perpendicular from the centre of curvature on the vehicle axis or, in other words, of the position of the vehicle on the track.

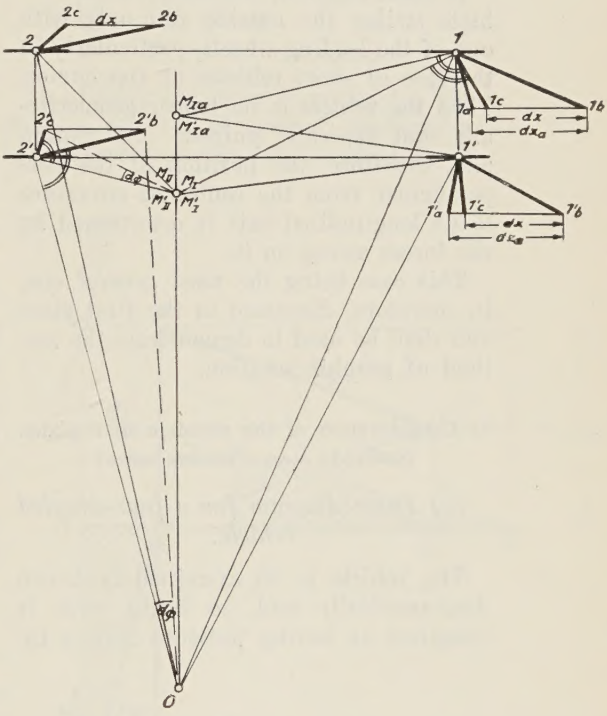


Fig. 1.

Assuming that the vehicle strikes the outer rail with the leading pair of wheels, the inner rail with the trailing pair of wheels, the distance p of the pair of wheels parallel through the centre of curvature from the striking leading pair of wheels, assuming a play σ between the

respective wheel flanges and the running edge of the rail, is given by :

$$p = \frac{l}{2} + \sigma \cdot \frac{R}{l} .$$

It often happens, however, that the vehicle strikes the outside rail only with one of the leading wheels, particularly in the case of short vehicles or flat curves. Then the vehicle is no longer geometrically, but statically guided. Its motion and, therefore, the position of the perpendicular from the centre of curvature to its longitudinal axis is determined by the forces acting on it.

This case being the most general one, is, therefore, discussed in the first place and shall be used to demonstrate the method of graphic solution.

1) Consideration of the problem in a plane.
(Guiding by means of pressure rollers.)

(a) Force diagram for a four-wheeled vehicle.

The vehicle to be examined is drawn diagramatically and, to begin with is imagined as having pressure rollers instead of flanges (2) as shown in figure 1*a*.

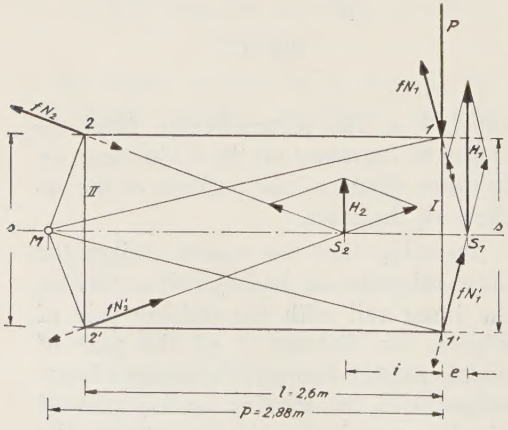


Fig. 1*a*.

During the slip rotation of the vehicle round a centre of friction M, the position of which is at first only assumed, the wheels slip perpendicularly to the connecting lines 1M, 1'M, 2M, and 2'M, i. e. the wheel contact points move momentarily in the direction of the dotted arrows and produce frictional forces fN_1, fN'_1, etc., acting in exactly opposite direction. At a non-striking pair of wheels the frictional forces between the rails and the two wheels, and at a striking pair of wheels the same frictional forces and in addition the directional force in the force diagram of the axle will result in an unbalanced transverse horizontal force H_I, H_{II}, H_{III}, etc., and in a moment $\mathfrak{M}$, which are transmitted by the axle guides or the bearing spring suspension to the frame. In the case of a vehicle with coupled pairs of wheels there are also two small longitudinal forces which are transmitted by the coupling rods to the other wheels (Strictly speaking there should also be considered in the case of each single axle a relatively small driving moment, or a small driving force to overcome the rolling resistance).

The vehicle is guided by a horizontal force, the guiding force P, which is transmitted by the outside rail, perpendicularly to the longitudinal axis of the vehicle, to the pressure roller of the striking leading wheel, (if the very slightly oblique position caused by the striking angle is disregarded).

In order that the vehicle shall be in equilibrium, all acting forces when added up in the force diagram must form a closed force-polygon, and in the plan of

(2) BÄSELER : « Spurerweiterung oder nicht » (Gauge widening or not). *Z.V.D.E.V.*, 1926, Nos. 8-10 and 12-13.

location the moment about any point, for example the point 1, must be zero.

The most simply imagined case for a vehicle statically guided in a curve, is a vehicle with a leading guiding pair of wheels, having equal wheel loads, i. e. equal to Q and not subject to any outside forces, the centrifugal forces being just balanced by the superelevation of the track.

We will ascertain the distance p of the centre of friction M from the front axle. M must be located on the longitudinal axis of the vehicle, because all wheel loads are equal to Q and the wheel tyres have been assumed as cylindrical and of equal tread circle radius. In addition M must be located a fraction behind the trailing pair of wheels because otherwise H_2 would not be able to balance the forces P and H_1 (fig. 1*a*).

A distance p of the front axle from M is assumed, and for this assumed position of M first the frictional forces of axle II and then those for axle I are combined for each axle separately, i. e. a force diagram is drawn as shown in figure 1*b*.

The two frictional forces fN_2 and fN'_2 result in a transverse force H_2 which acts at the point S_2 (figure 1*a*) and has a vertical outwards direction. The two frictional forces fN_1 and fN'_1 combine at the point S_1 with the transverse force H_1.

The two transverse forces H_1 and H_2 must balance the guiding force P, i. e., when combined they must be equal to P, and fulfil the condition :

$$H_1 \cdot e = H_2 \cdot i \text{ (fig. 1}a\text{).}$$

If p is at first assumed too small it will be found that $H_1 \cdot e < H_2 \cdot i$ and one has to try with a larger p, with the possible result that $H_1 \cdot e > H_2 \cdot i$. The correct p will be found by interpolation between these two values. For the example chosen (vehicle with a wheel base $l = 2.60$ m. and $f = 0.25$) this method gives : $p = 2.875$ m., and further :

$$H_2 = 0.175 \text{ Q}$$
$$H_1 = 0.49 \text{ Q and}$$
$$P = H_1 + H_2 = 0.665 \text{ Q.}$$

In addition the force diagram (fig. 1*b*) gives a relation of general application. As all forces on the vehicle must be in

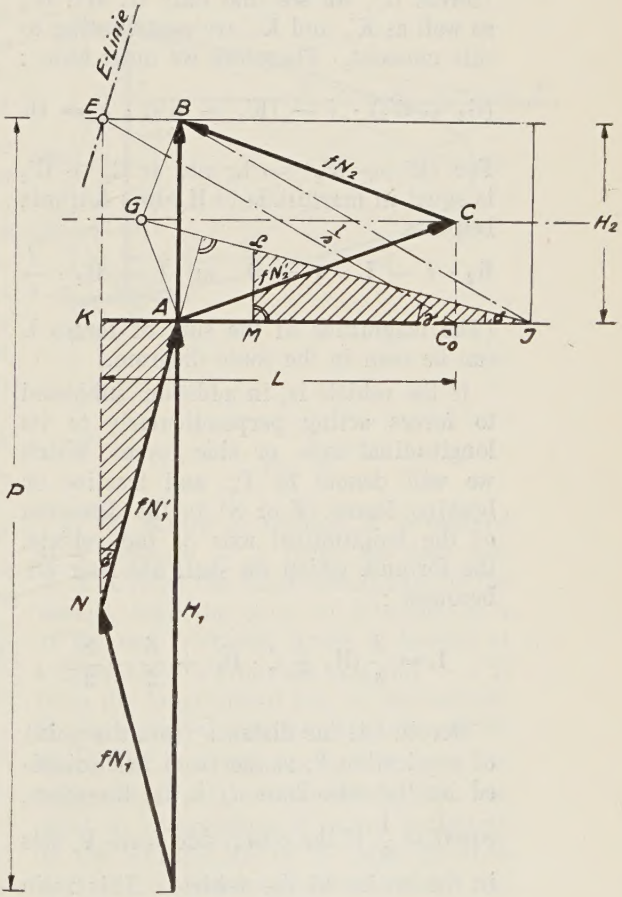


Fig. 1*b*.

equilibrium, the moment of all forces acting on the vehicle when projected in the rail plane must be equal to 0, for any point in this plane. Therefore, the moment round point 1 (see fig. 1*a*) upon which fN_1, and P act directly must be $= 0$; in other words, the moment of the forces fN_2, fN''_2 and fN''_1 each of which is equal to fQ, must be $= 0$ round point 1. If these three frictional forces are split into their components parallel to the longitudinal axis of the vehicle (forces K) and perpendicular thereto (forces G), we see that only G_2 and G'_2 as well as K'_1 and K'_2 are contributing to this moment. Therefore we must have :

$$(G_2 + G'_2) \cdot l - (K'_1 + K'_2) \cdot s = 0.$$

For $(K'_1 + K'_2) = L$, and as $G_2 + G'_2$ is equal in magnitude to H_2, this formula becomes

$$H_2 \cdot l - L \cdot s = 0 \quad \text{or} \quad L = H_2 \cdot \frac{l}{s}$$

(The magnitude of the sum of forces L can be seen in the force diagram.)

If the vehicle is, in addition, subjected to forces acting perpendicularly to its longitudinal axis, or side forces, which we will denote by P_s, and tractive or braking forces (Z or S) in the direction of the longitudinal axis of the vehicle, the formula which we shall use later on, becomes :

$$L = \frac{l}{s}(H_2 \pm \zeta \cdot P_s) + \frac{Z}{2} - \frac{S}{2}.$$

Herein ζ is the distance from the point of application P_s to the front axle, divided by the wheelbase l; it is, therefore, equal to $\frac{1}{2}$ if the outer side force P_s acts in the centre of the vehicle. This ratio facilitates a considerable simplification of the solution in nearly all cases. In the form

$$L = \left(\frac{l}{s}\right) \cdot H_2,$$

it will be seen that the line EC_0 (fig. 1*b*) must have the slope $1 : 1\frac{l}{s}$. E is the point of intersection of the horizontal through B with the vertical through the end point N of the force fN'_1, and C_0 is the point of intersection of the horizontal through A and the vertical through C. Starting, therefore, from the point C in the force diagram with an assumed value H_2, the line EC_0, having a known slope, is a locus of the point E. The second locus for E is the « E-line », a portion of which is given as a chain-dotted line in figure 1*b*, and which is shown as line

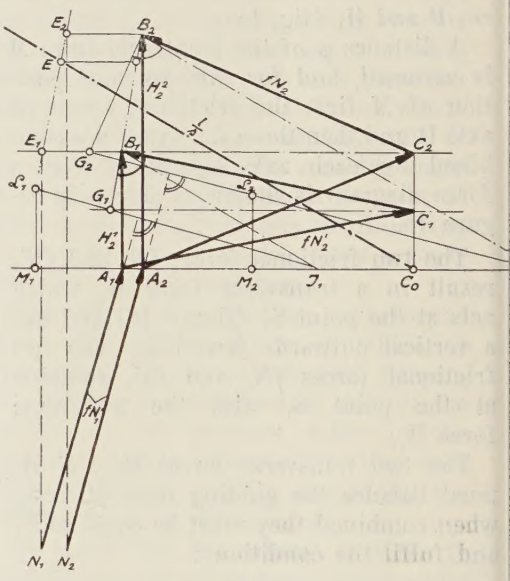


Fig. 1c.

E_1, E_2, in figure 1*c*. In this figure E_1 and E_2 are the corresponding points of intersection for two different force diagrams, for two differently assumed cen-

tres of friction M, and therefore also for two different forces H_2^1 and H_2^2 (H_2^1 smaller and H_2^2 larger than H_2), these two diagrams being superposed in such a way that the vertices C_1 and C_2 of the force triangles A_1, B_1, C_1 and A_2, B_2, C_2, lie in one vertical line above C_0. (Only the top portion of the force diagrams is shown). E_1 and E_2 are the points of intersection of the lines drawn horizontally through the end points B_1 and B_2 of the forces H_2^1 and H_2^2 in the force diagram (fig. 1*c*), with the verticals over N_1 and N_2, the end points of the force fN'_1, the direction of which is determined by the situation of the centre of friction. In figure 1*b*, E and N are to the left of A by the distance $\overline{A - K} = fN'_1 \sin \delta = K'_1$.

The point N in figure 1*b* is found in the following way : Draw the parallel to EC_0 through B up to J; then the points 2′, 2, and 1′, of the vehicle will correspond to the points A, B, J, in the correct ratio of length to width. To the centre of friction corresponds G, as the point of intersection of the perpendicular to AC at A, with the perpendicular on BC at B. The force AN must therefore be perpendicular to GJ, and by assumption, be equal to $fN'_1 = fQ$.

The distance $\overline{BE} = \overline{AK}$ is found more easily in setting off $\overline{JL} = fQ$ on JG and dropping from L the perpendicular LM on C_0A; then $\overline{LM} = \overline{BE}$. To obtain point E it is therefore only necessary to set off the distance $\overline{LM}$ to the left of B. In figure 1*c* the construction of the line E_1, E_2 is shown on a larger scale, viz. for the case $H_2^1 = 0.1\ Q$ and $H_2^2 = 0.2\ Q$. It will be seen that $\overline{L_1M_1} = \overline{B_1E_1}$ and $\overline{L_2M_2} = \overline{B_2E_2}$ are approximately equal. This means that the direction of the frictional forces on wheel 1′ is only slightly altered as H varies. When this method is applied to the given example, the same values as above are obtained for H_2, H_1 and P.

(b) Relations between the location of the centre of friction and the corresponding point of intersection S of the frictional forces.

The following simple relations exist between the location of the centre of friction M of an axle and the point of intersection S of the wheel friction forces (fig. 2) :

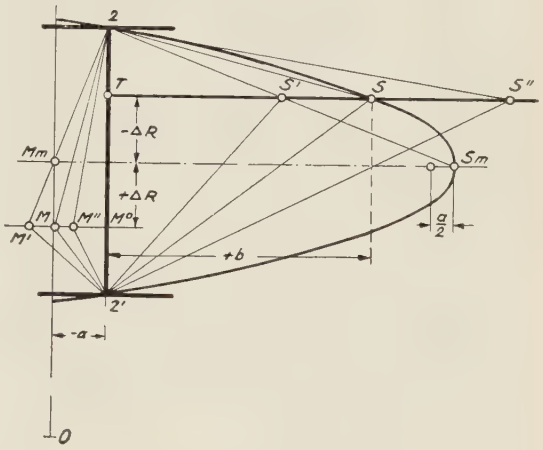


Fig. 2.

If the centre of friction M is selected at a distance — *a* from the axle and + Δ R from the longitudinal axis of the vehicle, then the point of intersection S of the two frictional forces is located at a distance + *b* from the axle and — Δ R from the longitudinal axis of the vehicle. If the centre of friction M moves on the line $\overline{M'\ M''}$ at a distance + Δ R from the longitudinal axis of the vehicle, then the point of intersection S moves infinitely on the straight line S′ S″ at a distance — Δ R from the longitudinal axis of the vehicle.

If the centre of friction M moves on the line OM at a distance — a from the axle, the point of intersection S moves on a parabola the vertex of which is located at S_m, which is the point of intersection relating to M_m located on the longitudinal axis of the vehicle, the arms of this parabola going through the points 2 and 2′; as a matter of fact, owing to the similarity of the triangles M M_0 2′ and 2′ TS, we have :

$$\frac{a}{\frac{s}{2} - \Delta R} = \frac{\frac{s}{2} + \Delta R}{b}$$

whence

$$a \cdot b = \frac{s^2}{4} - (\Delta R)^2$$

or

$$(\Delta R)^2 = \frac{s^2}{4} - a \cdot b$$

where a is constant and b is variable. This is the general equation of a parabola. By making the axis X pass through the vertex of the parabola the equation of the latter becomes :

$$(\Delta R)^2 = a \cdot x.$$

The parameter is $\frac{a}{2}$, the vertex lies at a distance $\frac{s^2}{4a}$ from the axis.

In practice the conditions for a vehicle running through a curve are not so simple as assumed until now. In the first place, it would be pure coincidence if all the wheel bearing pressures were equal. As is known, the centrifugal force alone already causes an alteration of the bearing pressures not to mention air resistance and other side forces. But even without such outside forces the sliding rotational motion alone and the H forces deriving therefrom cause an alteration of the bearing forces N. The transverse forces H (H_1, H_2 . . . H_i) arise at the upper edge of the rail head as the frictional forces originate on the rail. To overcome these H forces starting from the striking point, forces of equal magnitude must be transmitted through the striking axles to the frame and through the latter to the non-striking axles, unless H_1 or generally speaking an H force of a striking axle is concerned. Therefore, as shown in figure 3, a couple of

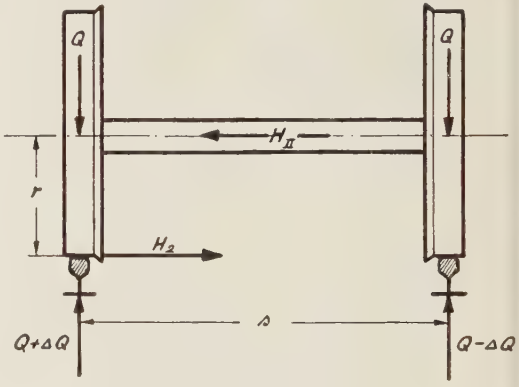


Fig. 3.

forces $H_2 \cdot r = H_{II} \cdot r$ arises at each non-striking axle, which causes an opposing couple of forces $\Delta Q \cdot s$. ΔQ is then $= \frac{H \cdot r}{s}$. If for example $r = 0.6$ metre, then the bearing pressure of the wheel towards which is directed the transverse force H_2 in the plane of the rail is decreased by $\Delta Q = H_2 \cdot \frac{0.6}{1.5} = \frac{2}{5} H_2$. The bearing pressure of the opposite wheel is increased by the same amount $\frac{2}{5} H_2$.

In addition, in actual fact the pressure rollers are replaced by wheel flanges

whereby special striking and contact conditions occur at the striking wheel. When taking these influences into account, the problem can no longer be treated as only « in a plane », but must be investigated by means of the conditions of equilibrium in space. As, however, in this case the forces in space stand in a simple relation to the forces in the horizontal plane, the graphical method can easily be employed for the analysis « in space ». In the following the above-mentioned influences will, therefore, also be considered.

2) Consideration of the problem in space.

The complete force diagram of the vehicle consists of the individual force diagrams for the different pairs of wheels which have a mutual influence on each other. We shall consider one after another the force diagrams of the fixed non-striking, striking, and coupled axles.

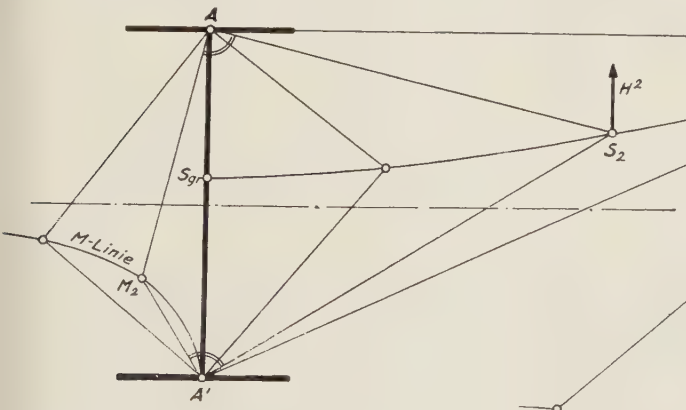


Fig. 4.

(a) The force diagrams of the rigid non-striking-axle.

For a given H (say H^2) the diagram of the axle forces can be drawn directly according to figure 4*a*, by setting off $H^2 = A_2B_2$ in magnitude, then drawing a circle around A as centre with a radius $f(Q - \frac{r}{s} \cdot H^2) = fN'_2$, this circle being intersected at C_2 by another circle with a radius $f(Q - \frac{r}{s} \cdot H^2) = fN_2$, with B as its centre. If parallels to B_2C_2 and A_2C_2 are now drawn in the location diagram through A and A', they will intersect at S_2 the point of application of H^2.

If perpendiculars on AS_2 and $A'S_2$ are erected at A and at A', these will intersect at M_2, the centre of friction for the axle, and to this point M_2 will correspond a point E_2 in the force diagram, as the point of intersection of the perpendiculars on B_2C_2 and A_2C_2 at B_2 and A_2. If the force triangles formed by H, fN' and fN are drawn for different H so that the vertices C are superposed, and that consequently C will move on the perpendicular line C_0-C_3-C_{gr}, B will move on the

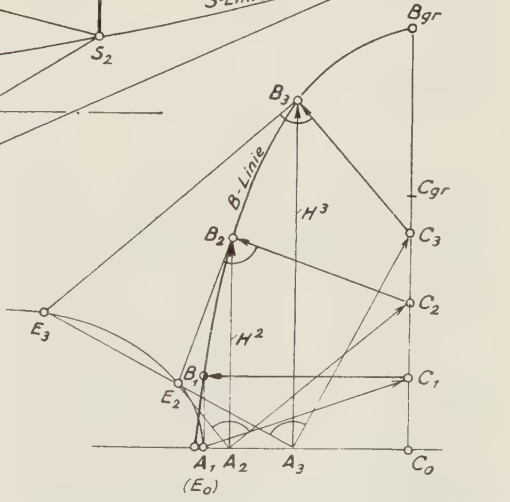


Fig. 4*a*.

so called B-line, E on the curve E_0-E_3, M forwards from A′ on the M-line, which is the locus of the centre of friction when H varies, and S on the curve S_1-S_{gr}. With a very small H, when $H < 4 f^2 \frac{r}{s} Q$, that is with $f = 0.25$, $r = 0.60$., smaller than :

$$4 \cdot \frac{1}{16} \cdot \frac{0,6}{1,5} \cdot Q = \frac{Q}{10}$$

the centre of friction will remain at A′ because, should it pass point A′, the direction of the frictional force fN' acting at A would be reversed so that it could no longer combine with the frictional forces fN and H in a closed force diagram (unless there arose a very large additional braking force). The frictional force at point A′ is no longer too large, to be overcome at A′; it becomes inherent friction and takes on an indefinite value smaller than $f\ (Q + \frac{r}{s} \cdot H)$, but the centre of friction of the axle remains fixed at A′, i. e. the axle rotates around the wheel A′. Point S moves from S_1 on the horizontal through A to the right into the infinite.

All forces arising between the rails and the two wheels of the axle can be represented through the single transverse force H at point S.

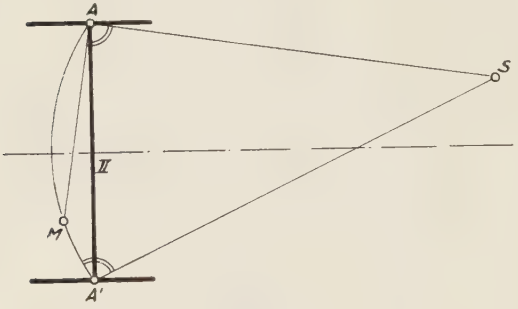


Fig. 5.

(b) The force diagram of the non-striking coupled axle (called axle II).

On account of being coupled with other axles, the frictional forces of the wheels must not only take up a horizontal force H_{II} and a moment $\mathfrak{M}_{II}$, but in addition must also take up a small longitudinal force, because of the compulsory equal rotation of coupled wheels; this longitudinal force does not only vary with H and N, but also with the transmission of the force from the remaining wheels to the coupled axles. Assuming a given constant H_2 selected to be set off as the beginning of the force diagram under the form A A_0 (fig. 5*a*), the vertex C of the triangle of forces A C B can adopt many positions varying with the magnitude of the longitudinal force to be transmitted from the other coupled axles to the axle considered. Now, all these points C are located on the arc of circle $C_u C^0$ drawn around A as a centre with a radius $f\ (Q + \frac{r}{s} H_2) = fN'_2$, whilst B may move from B_r to B_l, but C_u B_r must always equal $f\ (Q - \frac{r}{s} \cdot H_2) = fN_2$. as far as

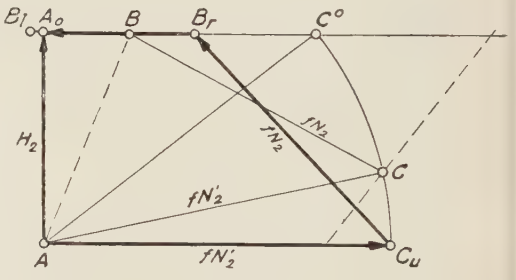


Fig. 5*a*.

C^0B_1. The centre of friction M in the location diagram, figure 5, moves then from A via M to A'. The segment A_0B represents, in the force diagram, the magnitude of the longitudinal force to be taken over by the other axles. The centre of friction is at first fixed at the points A' and A, and does not move beyond them unless specially large tractive forces Z or braking forces S cause a lateral displacement of the centre of friction as will be seen later.

(c) The force diagram of the striking axle.

The outside wheel of the front axle will always strike in a curve; we shall therefore consider a front axle I as an example of a striking axle. Let us assume that the alteration of the load due to the influences of outside forces will give a wheel load Q_1 on the left striking wheel and a wheel load Q'_1 on the right wheel; if no other external forces, such as external lateral forces, tractive forces, braking forces, are present, then $Q_1 = Q'_1 = Q$. When running through a curve the influence of the other axles on the same vehicle frame are added thereto, and also the forces H exerted on them by the frame at the level of the centre line of the wheel set, thus altering the wheel loads. The forces H_{II}, H_{III}... (see table I the for indices of H) of the non-striking axles (with a four-wheeled vehicle the force $-H_{II} = +H_2$), with which the axle guards of the axles II, III... press against the axles II, III..., arise at the rail pressure point $\mathfrak{A}$ and are an essential factor (guiding pressure) of the guiding force P. This is clear as the frame must also be in equilibrium.

If no outside forces are assumed, the rail presses with the force $H_{II} + H_{III}$... against the striking wheel (1), and the axle (I), against the axle guard of the striking wheel (1). Consequently if $H_{II} = -H_2$, $H_{III} = -H_3$, etc., the striking wheel (1) presses with the force $H_2 + H_3 ... = H_I$ against the guiding rail. The striking left side wheel (1) receives an increase in pressure $\Delta Q = \frac{r}{s} H_I$. The frictional force at wheel (1') becomes $fN'_1 = f(Q - \frac{r}{s} H_I) = \frac{Q}{4} - \frac{1}{10} H_2$ for $f = \frac{1}{4}$, $H_I = H_2$ and $r = 0.60$ m., and has the same magnitude as the frictional force on wheel (2). The vertical pressure

$$Q + \frac{r}{s} \cdot H_I = (Q + \frac{2}{5} \cdot H_I)$$

acts on wheel (1); the rail must produce a counter pressure $Q + \frac{r}{s} \cdot H_I$ in the upwards direction. Assuming a two-point contact, this pressure will be distributed over two points of the striking wheel (1), because wheel (1) will then be forced against the head and side of the rail. The almost vertical pressure N_1 on the surface of the rail head is not equal to $Q + \frac{r}{s} \cdot H_I$, but considerably smaller, because the flange pressure point $\mathfrak{A}$ takes over a part of the vertical force $(Q + \frac{r}{s} \cdot H_I)$.

The distance a by which the flange pressure point $\mathfrak{A}$ lies below the rail head surface must be determined separately. For the purpose of the present analysis, we will assume it to be 8 mm. (5/16") below the upper edge of the rail, which is approximately the case for cylindrical wheel tyres. It is not vertically below the wheel contact point A_1, but ahead of this point by the amount b. The lead is calculated from the striking angle α and

the wheel radius r, and according to BÄSELER's paper « Die Spurkranzreibung » (Flange friction) : $b = (r + a) \cdot \alpha \tan 60^0$ (assuming that $\alpha = \sin \alpha$).

At the flange pressure point $\mathfrak{A}$ which, with two-point contact is located on the flange cone, the force $\mathfrak{N}$ acts from the rail against the wheel perpendicularly to the face of the wheel flange. This force

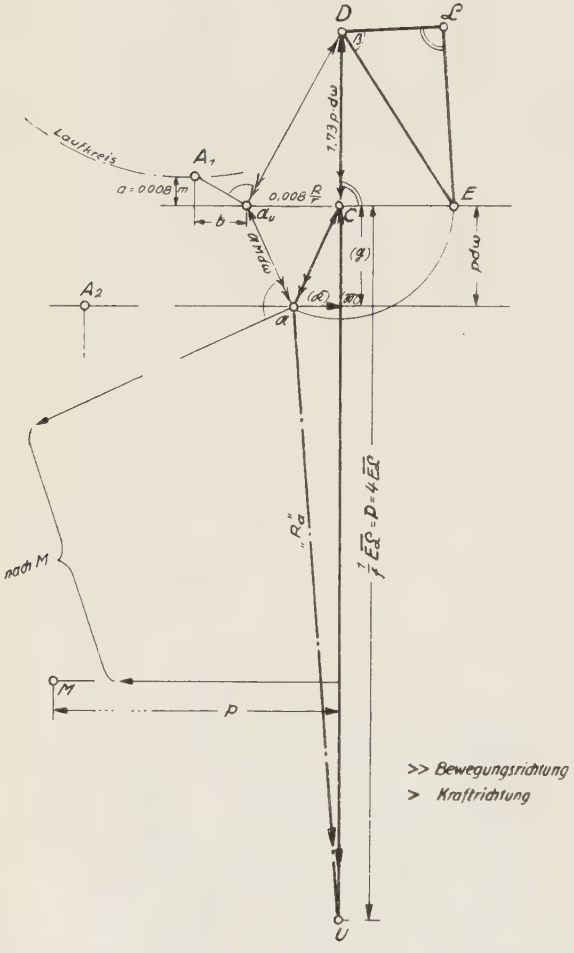


Fig. 6.

Note. — Bewegungsrichtung = direction of motion. — Kraftrichtung = direction of force. — Laufkreis = tread circle. — Nach M = to M.

$\mathfrak{N}_y$ is split into $\mathfrak{N}_y = P = \mathfrak{N} \sin \beta$ and $\mathfrak{N}_z = \mathfrak{N} \cos \beta$. The magnitude of the frictional force at point $\mathfrak{A}$ is $f \cdot \mathfrak{N}$.

The determination of the direction of $f \cdot \mathfrak{N}$ is a problem of descriptive geometry.

Point $\mathfrak{A}$ rotates, while sliding, on the one hand with the complete vehicle in the horizontal plane around the centre M of the frictional forces exerted at points A_1 and A'_1, and, on the other hand, on account of the rotation of the wheel in the vertical plane, it also rotates around a horizontal transverse axis going through wheel contact point A_1 (fig. 6).

When the vehicle performs an infinitely small rotation represented by angle $d\omega$ ($d\omega$ is much enlarged = drawn $\frac{1}{100}$) in the horizontal plane, the point $\mathfrak{A}$ moves perpendicularly to line M$\mathfrak{A}$ from $\mathfrak{A}_u$ to $\mathfrak{A}$ by an amount $\overline{\mathfrak{A}\,\mathrm{M}} \cdot d\omega$, and if $d\omega$ is drawn $= \frac{1}{100}$ for reasons of simple and clear representation, then $\mathfrak{A}_u$ is located at a distance p from the horizontal line $\mathfrak{A}$ ($\mathfrak{M}$) with the meaning given above, but set off in cm. instead of in metres.

Figure 6 represents, below the line $\mathfrak{A}_u$ CE, motions and forces located in the horizontal plane. Point $\mathfrak{A}$ moves perpendicularly to line $A_1\,\mathfrak{A}_u$ in the vertical plane located above line $\mathfrak{A}_u$ CE which plane should be imagined to be bent around $\mathfrak{A}_u$ CE towards the top of the figure, and as the rotation of the wheel is $\frac{R}{r}$ times faster than that of the vehicle, the magnitude of the motion, assuming $r + a = r$, i. e. the length $= \overline{D\,\mathfrak{A}_u}$ $(a \overset{\wedge}{+} r \cdot \sin \alpha \cdot \operatorname{tg} \beta)\ \frac{R}{r} \cdot d\omega = a \overset{\wedge}{+} \frac{r}{R} \cdot p \cdot \operatorname{tg} \beta) \cdot \frac{R}{r} \cdot d\omega$, is, for $d\omega = \frac{1}{100}$, $a = 0\ 008$ m.,

p in metres, $\tan \beta = 1.73$, $= (0.008 \overset{\wedge}{+} \frac{r}{R} \cdot p \cdot 1.73) \cdot \frac{R}{r}$ in centimetres.

Therefore, it is enough to set off from $\mathfrak{A}_u$ the constant value $0.008 \frac{R}{r}$ cm. (with $R = 200$ m. and $r = 0.60$ m. $= 2.667$ cm.) to the right up to point C and to go upwards from the new point C by an amount p (in metres). 1.73 (in our case with p assumed $= 2.90$ m., by $2.90 \cdot 173 = 5$ cm.) up to D. Then $D\mathfrak{A}_u$ is the motion corresponding to $\overline{\mathfrak{A}_u \mathfrak{A}}$ in the vertical plane. Considered in space, point $\mathfrak{A}$ moves accordingly from point D, raised again into the vertical plane, to $\mathfrak{A}$. $\overline{D\mathfrak{A}}$ (seen in space with upwards bent vertical plane) is the magnitude of the complete sliding path of point $\mathfrak{A}$, $\overline{C\mathfrak{A}}$ represents the magnitude of the horizontal projection of $\overline{D\mathfrak{A}}$, and $\overline{\mathfrak{M}(\mathfrak{A})}$ is the magnitude of the sliding path in the X direction, $\overline{C(\mathfrak{M})}$ the magnitude of the sliding path in the Y direction, and $\overline{DC}$ the magnitude of the sliding path in the Z direction.

To obtain the magnitude of the total sliding (on one segment), a circle is drawn around C as centre with $C\mathfrak{A}$ as radius, up to E, and the magnitude of the value of the complete sliding path of the point $\mathfrak{A}$ of the flange is obtained at $\overline{DE}$.

The frictional force $f\mathfrak{N}$ at point $\mathfrak{A}$ moves in the opposite direction to the sliding path of point $\mathfrak{A}$; its projections on the three planes are the same as those of the total sliding path of point $\mathfrak{A}$, only they are in the opposite direction.

Assuming the total frictional force $f\mathfrak{N} = \overline{DE}$, then the component in the direction of the longitudinal axis of the vehicle (X-direction) will be $\mathfrak{K} = \overline{\mathfrak{A}(\mathfrak{M})}$, the component in the direction of the wheel axle (Y-direction) will be $\mathfrak{G} = \overline{C(\mathfrak{M})}$; $\mathfrak{G} \overset{\wedge}{+} \mathfrak{K} = \overline{\mathfrak{A}C}$, the vertical component $\mathfrak{F} = \overline{CD}$. The magnitude of the force $\mathfrak{N}$ is $\frac{1}{f} \cdot DE$. Assuming $\mathfrak{N}$ for approximation, as located in a vertical plane parallel to the axis of rotation of the pair of wheels, and inclined in this plane by an angle of $(90° - \beta) = 30°$ on the horizontal, the right-angled triangle DLE can be drawn with $\overline{DE}$ as hypotenuse, and with the adjacent angles β and $(90°\text{-}\beta)$, so obtaining $\mathfrak{N}_y = P = \left(\frac{1}{f}\right) \cdot \overline{EL}$, and similarly, $\mathfrak{N}_z = \left(\frac{1}{f}\right) \overline{DL}$.

In the plan (fig. 6) drawn at a 1 : 10 scale, only the horizontal projections of the forces appear. When setting off from $\overline{\mathfrak{A}C}$, $\mathfrak{A}$ in length and direction, further $\left(\frac{1}{f}\right) \overline{EL}$ in magnitude from C perpendicularly downwards to U, neglecting the small angle α, the magnitude and direction of the forces acting at point $\mathfrak{A}$ in the horizontal plane are given at $\overline{\mathfrak{A}U}$, hence the direction of the resultant from the guiding force P and the frictional forces $\mathfrak{G}$ and $\mathfrak{K}$ at point $\mathfrak{A}$. This resultant force shall be called henceforth R_a and is shown in figure 6 as a chain-dotted line —.—.

Thus the direction of all forces acting on the striking wheel at points A_1 and $\mathfrak{A}$ are known, as well as the magnitude and direction of the frictional force fN'_1 at point A'_1, on the non-striking wheel. The forces need only be combined as to magnitude and direction in accordance with static laws, in order to obtain also the magnitude of the forces on the striking wheel in the force diagram; because, if all forces exerted on the axle are to be

in equilibrium, the forces projected on the horizontal plane must also be in equilibrium in this plane.

If the centre of friction has been selected correctly, not only must the corresponding force diagram form a closed force polygon in the horizontal plane, but the sum of the vertical forces must also be zero.

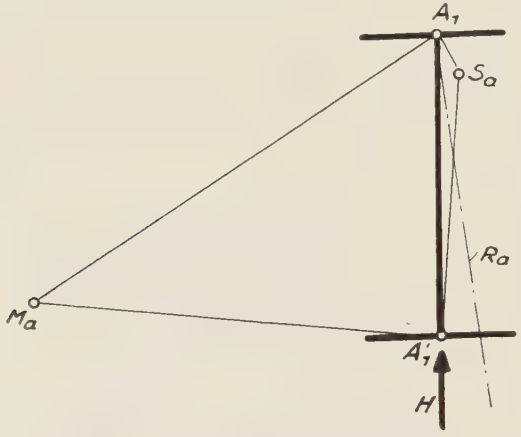


Fig. 7.

Let us assume the correct centre of friction to be at point M_a (fig. 7). Only the force $H_I = \overline{LO}$ transmitted from the frame of the vehicle acts on the axle in its direction, and, of course, the wheel pressures Q_1 and Q'_1, which can be calculated from H_1. The force polygon (fig. 7*a*) is now drawn, by setting off, from point L to point N, the force $fQ'_1 = fN'_1 = \frac{1}{4} N'_1$ parallel to $A'_1 S_a$ of fig. 7, and by drawing from N the parallel to $A_1 S_a$, and setting off from point O up to the point of intersection U the direction of the resultant R_a from the forces ⅁, ℜ and P, this direction being ascertained by the method described above. $\overline{NU}$ must then have the value $f \cdot N_1 = \frac{1}{4} N_1$; the magnitude of the pressure on the rail head at

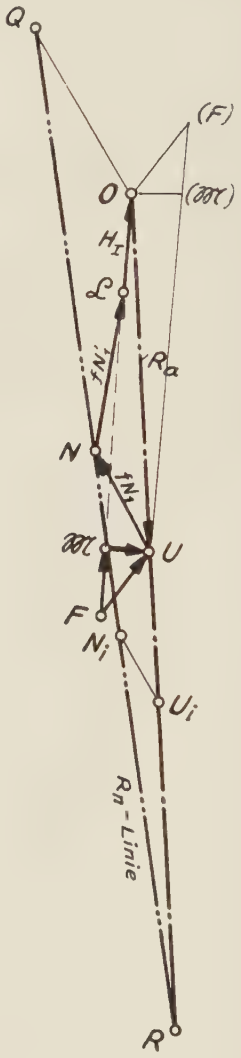


Fig. 7*a*.

point A_1 is therefore given by $N_1 = \frac{1}{f} \overline{NU}$.

Finally, the condition that the sum of

the vertical forces at points A_1 and $\mathfrak{A}$ (i. e. on wheel 1) $= Q_1$, remains to be fulfilled. At point A_1, only the force N_1 acts vertically upwards. At point $\mathfrak{A}$, the vertical projection of the flange pressure $\mathfrak{N}$ is $\mathfrak{N}_z$, and the vertical projection of the frictional force $= \mathfrak{F}$. Therefore : $N_1 + \mathfrak{N}_z + \mathfrak{F}$ must be $= Q_1$. Both forces $\mathfrak{N}_z$ and $\mathfrak{F}$ are contained in the force diagram, without being visible. With $\overline{OU}$ (fig. 7*a*) as a basis, lines drawn parallel to the directions of forces found according to figure 6 will result in a diagram $\overline{OU}$ ($\mathfrak{M}$) (F) which is in figure 7*a* similar to diagram $\mathfrak{A}$ U ($\mathfrak{M}$) C in figure 6, or to the construction U O $\mathfrak{M}$ F, rotated by 180°.

In this diagram $\overline{U\ (F)} = \overline{OF}$ corresponds to the force $\mathfrak{N}_y = P$, as it is equal to $\overline{UC}$ in figure 6, and $\overline{F\,\mathfrak{M}}$ corresponds to the force $\mathfrak{S}$, as it is equal to C ($\mathfrak{M}$) in figure 6. $\mathfrak{N}_z$, which we seek, is however $\mathfrak{N}_z = \mathfrak{N}_y \operatorname{cotan} \beta = 0.577\ \overline{OF}$ (when $\beta = 60°$). (With a scale, for the forces, of 1 dm. $= \frac{1}{4}$ Q, $\mathfrak{N}_z$ will therefore be 0.145 $\overline{OF}$, measured in dm.) $\cdot\ \mathfrak{F} = \mathfrak{S} \cdot \tan \beta = \tan 60° \mathfrak{S} = 1.73\ \overline{F\,\mathfrak{M}}$. (With a scale, for the forces, of 1 dm. $= \frac{1}{4}$ Q, $\mathfrak{F}$ will therefore be 0.43 $\overline{F\,\mathfrak{M}}$ measured in dm.).

In order to have equilibrium :

$$\frac{1}{f}\overline{NU} + \operatorname{cotan}\beta \cdot \overline{OF} + \tan\beta \cdot \overline{F\mathfrak{M}} = Q_1.$$

$$\overline{NU} + \operatorname{cotan}\beta \cdot f \cdot \overline{OF} + \tan\beta \cdot f \cdot \overline{F\mathfrak{M}} = Q_1 \cdot f;$$

and for the assumed numerical values :

$$\overline{NU} + 0.145\ \overline{OF} + 0.43\ \overline{F\mathfrak{M}} = \frac{Q_1}{4}.$$

$$Q_1 = Q + \Delta Q.$$

Set off in the scale 1 cm. $= \frac{1}{40}$ Q, $fQ = \frac{Q}{4}$ is equal to 10 cm. whence $\overline{NU} + 0.145\ \overline{OF} + 0\ 43\ \overline{F\mathfrak{M}} = 10\left(1 + \frac{\Delta Q}{Q}\right)$ cm. $= Q_1 \cdot f$.

The total $0.145\ \overline{OF} + 0.43\ \overline{F\,\mathfrak{M}}$ is sometimes better replaced by the value $\xi \cdot \overline{OU}$, so that we get : $0.145 \cdot \overline{OF} + 0.43\ \overline{F\,\mathfrak{M}} = \xi \cdot \overline{OU}$.

It has actually been ascertained by a large number of tests and calculations that the value of ξ for a definite R remains nearly constant in the cases which generally occur ($p > 2.00$ m.); because it is visible from figure 6 that $\overline{UC}$, $\overline{C\ (\mathfrak{M})}$ and $\overline{\mathfrak{A}\ U}$, which correspond to the forces $\overline{OF}$, $\overline{F\mathfrak{M}}$ and $\overline{OU}$ in figure 7*a*, vary approximately proportionally to $p \cdot d\omega$, as well as $\overline{CD}$, $\overline{\mathfrak{A}\ C}$ and $\overline{EL}$. Therefore, in spite of the variations of p, the ratio of $\overline{OF} + \overline{F\,\mathfrak{M}}$ to $\overline{OU}$ remains nearly constant. For R = 200 m., for example, $\xi = 0.225$ to 0.245, the mean value being 0.235. ξ decreases with increasing curve radii. To a curve of R = 200 m. will therefore correspond :

$$\overline{NU} + 0.235\ \overline{OU} = (Q \pm \Delta Q) \cdot \frac{1}{4}.$$

$$\overline{NU} + \xi \cdot \overline{OU} = 10\ (1 + \frac{\Delta Q}{Q}) \text{ cm.}$$

(It is only in cases of very small values of p that the value $\xi = 0.235$ does not apply.) The graphical representation of this formula is as follows :

If one sets off at O the distance $\overline{OQ} = fQ_1 = \frac{Q_1}{4}$, parallel to A_1S_a in figure 7, and therefore also parallel to NU, and if $\overline{OR} = \frac{1}{\xi} \cdot \overline{OQ} = \frac{1}{0.235} \cdot \overline{OQ} = 4.25\ \overline{OQ}$,

point N must be located on the line joining Q and R, i. e. the line QR —..—..— is a locus for N. In other words, if N is located on line QR, the condition : The sum of the vertical forces round points A_1 and $\mathfrak{A} = 0$ is fulfilled. Proof : Every parallel to fN_1 through any one point N_i of the line QR cuts off on the line OR a distance $\overline{OU_i}$ which fulfils the required condition

$$\overline{N_iU_i} + \xi \cdot \overline{OU_i} = (Q + \Delta Q) f = \overline{OQ}$$

because we have :

$$\frac{\overline{N_iU_i}}{\overline{U_iR}} = \frac{\overline{QO}}{\overline{OR}} = \xi$$

$$\overline{N_iU_i} = \xi \cdot \overline{U_iR}$$

$$\overline{N_iU_i} + \xi \cdot \overline{OU_i} = \xi \cdot \overline{U_iR} + \xi \cdot \overline{OU_i} = \xi \cdot \overline{OR} = \overline{QO}.$$

The direction of the line QR shall be called henceforth R_n and shown as a chain-double-dot line —..—..—...

In order to find, when solving these problems, the directions of both lines R_a and R_n more rapidly, and in order to ascertain the course of these two directions for different positions of M, i. e. for most diverse values of p and ΔR, for example $R = 200$ m., the different directions have been combined in figure 8 in which it is only necessary to take up the corresponding ΔR as abscissa, and the corresponding p as ordinate, and to join them with the origin O of the co-ordinates, in order to find in the lower portion of the figure, the direction of R_a (—.—.—.), and in the upper portion the direction of the R_n line (—..—..—..).

It may be seen that when the centre of friction moves outwards and consequently when ΔR decreases and becomes negative, the R_a line will be diverted quite steadily from the vertical to the right, whereas the R_n lines only very slightly change their direction at the centres of friction near the longitudinal axis of the vehicle and will only become steeper for large negative ΔR. This nomogram, only shown diagramatically in figure 8, has been drawn up based on a large number of completely worked-out examples. It is possible to draw up the nomogram to any desired scale; but care must be taken that the ordinates and abscissæ are in the right proportion, as the object is only to set off quickly the *directions* of R_a and R_n.

The object of the nomogram is to carry out quickly a great number of investigations for a given curve radius and a given type of vehicle, with approximately constant p, but variable ΔR, without being compelled to draw up the directions of the R_a and R_n lines every time.

To sum up, it may be said about the relations of forces exerted on the striking wheel that :

For a given centre of friction M, the directions of the frictional forces fN_1 and fN'_1 are found directly from the diagram of location, and furthermore the direction of the horizontal projection of the forces at $\mathfrak{A}$ by a simple geometrical protraction.

After combining the forces in the force diagram, the line QR gives in addition a criterion of the question as to whether the sum of the vertical forces on the axle is $= 0$.

The last condition, i. e. that the sum of the vertical forces shall be $= 0$, will rarely be fulfilled. It must in particular be remembered that the forces $H_2 \pm H_3 + \ldots = H_1$, to be transmitted from the other axles of the vehicle through

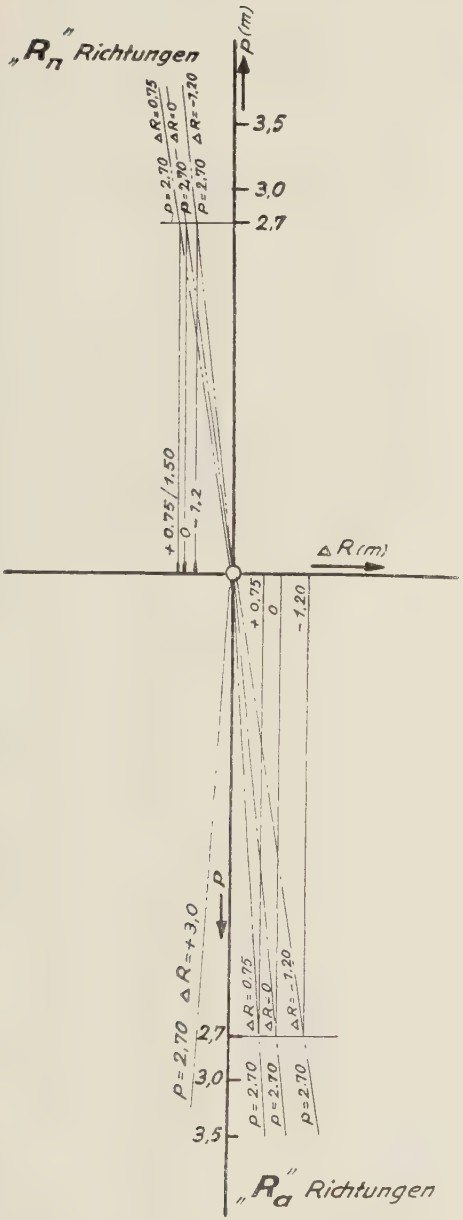


Fig. 8.

Note. — Richtungen = directions.

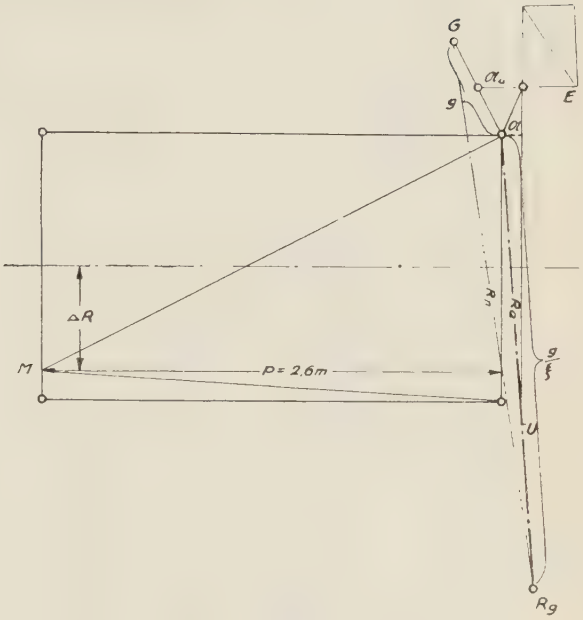


Fig. 9.

axle I and the striking wheel to the rail, alter the wheel pressures Q_1 and Q'_1, and therefore also the magnitudes of N_1 and N'_1 as well as of fN_1 and fN'_1. We shall therefore go one step further in our consideration of the axle I, and shall now discuss the question :

Which force H, for a given centre of friction of axle I (see fig. 9), can and must be transmitted to the rail through axle I in order that equilibrium shall exist ?

The direction of fN'_1 and fN_1 is given by the position of the centre of friction M, and furthermore the direction of $\mathfrak{U}$ can be derived from R_a as shown in figure 6*a*. Then the force diagram is drawn up.

As external force we shall assume a side force $P_s = \overline{LO_0}$ directed upwards and caused by say a high superelevation.

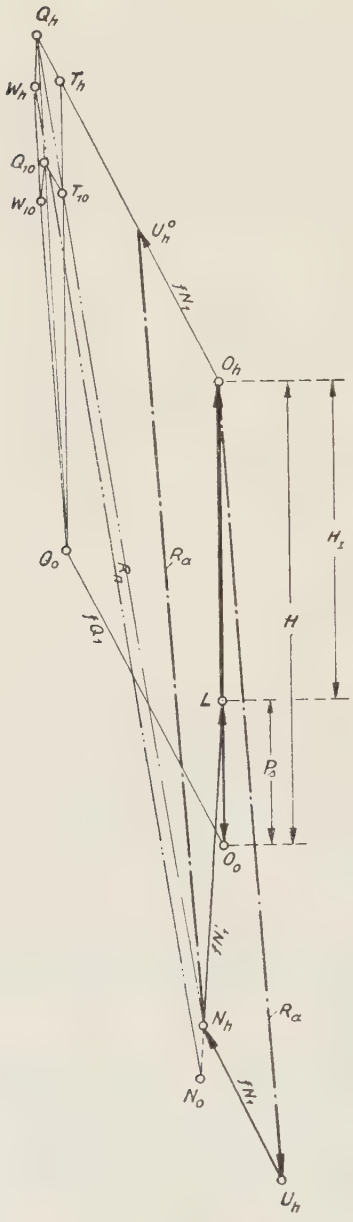


Fig. 9*a*.

The force $fN'_1 = \overline{LN_0}$ is added to force $\overline{LO_0}$ at L, and at 0_0 the force $fQ_1 = 0_0\ Q_0$, wherein Q_1 means the change in wheel pressure, resulting from P_s.

It is best to construct the direction of the chain-double-dot line R_n immediately in the plan of location (fig. 9) by setting off any distance $g = \overline{\mathfrak{A}\ G}$ on the line $\mathfrak{A}\,\mathfrak{A}_u$, and by protracting on the line $\mathfrak{A}\ U$ in the direction of R_a the distance $\frac{g}{\xi} = 4.25\ g$ up to R_g; GR_g is then the required direction of R_n. In order that equilibrium of the vertical forces shall exist, point N_0 in the force diagram, as in figure 7*a*, should be located on the parallel to GR_g through Q_0. This, however, is not the case. A force H must be added in order to balance the forces. If, however, axle I transmits a force H to the rail, the wheel load Q_1 is increased by $\Delta Q = \frac{r}{s} \cdot H$, and the wheel load Q'_1 is decreased by the same amount $\Delta Q = \frac{r}{s} \cdot H$, and accordingly fN'_1 decreases by $f \cdot \frac{r}{s} \cdot H$, i. e. in our case by $\frac{1}{4} \cdot \frac{0.6}{1.50} \cdot H = \frac{1}{10} \cdot$ In the same way fQ_1 increases by $\frac{1}{10} \cdot H$. When setting off, in the force diagram, a force H = 10 cm. vertically upwards from point Q_0 and parallel to LO_0, as far as T_{10}, then $\overline{O_0\ Q_0}$ will increase by $\frac{10}{10} =$ 1 cm. as far as Q_{10}, whereas $fN'_1 = LN_0$ will decrease by 1 cm. This one cm. is represented by the segment $\overline{W_{10}\ Q_{10}}$. Now we join Q_0 with W_{10}, Q_{10} and T_{10}. Then the line $Q_0\ Q_{10}$ will be the locus for the correct Q. In order to ascertain Q_h definitely it is enough to start from N_0, parallel to $R_g G$ to the point of intersection W_h, and from this point to draw a parallel to $N_0 L$ up to the point of inter-

section Q_h. A parallel to $Q_0\ O_0$ is drawn from Q_h as far as the point of intersection with the vertical O_0 L at O_h, and finally the parallel to GR_g up to the point of intersection N_h on the line LN_0. Distance $\overline{Q_0\ Q_h} = \overline{O_0\ O_h}$ provides the H force required for equilibrium, being equal to 13 cm. in the force diagram, which for a force scale 1 : 40 means $\frac{13}{40} \cdot Q = 0.325\ Q$. The force $fN_1 = \overline{LN_h}$ has become smaller than LN_0 by $\frac{13}{10} = 13$ cm., and the force $fQ_1 = \overline{O_h\ Q_h}$ has become 1.3 cm. larger than $\overline{O_0\ Q_0}$, and finally point N_h is located on the chain-double-dot line through Q_h, which ensures that the sum of the vertical forces acting on axle I is well 0. fN_1 is represented by a distance $\overline{N_h\ U_h} \cdot H_I = \Sigma\ (H_2 + H_3 + \ldots) - P_s = H - P_s$.

This construction enables one to ascertain graphically the H corresponding to any given ΔR and p. It should be noted that the magnitude of p, i. e. the distance of the centre of friction from the axle I, only plays a secondary part for ascertaining the H required for equilibrium. The magnitude of ΔR is, on the contrary, very important, so that it can almost be said that to a given ΔR corresponds a definite H, which is easily constructed by the method explained above. The static moment of the forces transmitted to the vehicle by the axle guards of axle I is equal to the component of fN'_1 in the X direction multiplied by s, i. e. $\mathfrak{M}_I = K'_1 \cdot s$, as all other forces in the plan go through point 1 (very approximately). The frictional force fN_1 is ascertained by setting off from O_h or from N_h the R_a direction, which cuts off $\overline{N_h\ U_h}$ or $\overline{O_h\ U^0_h} = fN_1$ (see fig. 9*a*).

3) Determination of the forces from the force diagrams.

The frictional forces in Q units on the rail head are obtained from the force diagrams by multiplying the respective force segments by the scale factor (here always $\frac{1}{40}$, for example).

The normal pressures of the wheel on the rail head are obtained in Q units, from the same force segments by reading them off in decimetres. The normal flange pressure $\mathfrak{N}$ is obtained, in Q units from the force diagram, by multiplying the segment $\overline{OF}$, representing P in cm. (fig. 7*a*) by $1/\sin\beta = 1.16$ and the force scale $\frac{1}{40}$, i. e. by $(1/\sin\beta) \cdot (1/40) = 0.029$.

The frictional energy A_R of a non-striking wheel is the frictional force multiplied by the path of this force in the force direction. The frictional force is obtained in Q units from the force diagrams by multiplying the respective force segment in cm. by the scale factor $\left(\frac{1}{40}\right)$. The distance of the path of friction is obtained from the diagram of location by reading off the distance d of the centre of friction from the contact point in dm. and multiplying it by φ, the angle of rotation of the whole vehicle, in other words, the length of the curve divided by R.

Therefore

$$A_R = fN \cdot (\overline{MA}) \cdot \varphi = fN \cdot d \cdot \varphi.$$

The frictional energy on a striking wheel is $AR + \mathfrak{A}_R$. $\mathfrak{A}_R$ is obtained from the force diagram by multiplying the flange pressure $\mathfrak{N}$ by $f\ (=\frac{1}{4})$ and by the length of the path of this frictional force. This path corresponds to the distance DE

in figure 6 and its ratio to the path of point 1 on the rail head is as $\overline{DE}$: $\overline{\mathfrak{A}\,\mathfrak{A}_\mu}$; it is therefore equal to

$$\frac{\overline{DE}}{\overline{\mathfrak{A}\,\mathfrak{A}_\mu}} \cdot d_1 \cdot \varphi.$$

$$A_{RI} + \mathfrak{A}_{RI} = (fN_1 \cdot d_1 + f\mathfrak{R}\,\frac{\overline{DE}}{\overline{\mathfrak{A}\,\mathfrak{A}_u}} \cdot d_1) \cdot \varphi.$$

The curve resistance W for an infinitely small distance $R \cdot d\varphi$ is given by :

$$W = \frac{d\varphi\,(\Sigma A_R + \Sigma \mathfrak{A}_R)}{R \cdot d\varphi} = \frac{\Sigma A_R + \Sigma \mathfrak{A}_R}{R}$$

$$W = \frac{1}{4\,R} \cdot (\Sigma N d + \Sigma\,\mathfrak{R}\,\frac{\overline{DE}}{\overline{\mathfrak{A}\,\mathfrak{A}_\mu}} \cdot d) \text{ in Q units.}$$

If the ratio $\frac{\overline{DE}}{\overline{\mathfrak{A}\,\mathfrak{A}_\mu}}$, is called η_1,

$$W = \frac{1}{4\,R}(\Sigma N \cdot d + \Sigma\,\mathfrak{R} \cdot \eta_1 \cdot d).$$

The specific curve resistance w becomes $w = \frac{W \cdot 1000}{n}$ kgr./t., in which n = number of wheels. Only the frictional energy $\mathfrak{A}_R$ arising at the striking wheels is of importance when considering the wear at the sides of the wheel, and is therefore of special interest.

D) General force diagram for vehicles with rigid axles.

We have shown in chapter C the aspect of the individual force diagrams, if the forces acting on the individual axles are to be in equilibrium. Now, the complete vehicle, including its axles, must also be in equilibrium, i. e. it must be possible to combine the force diagrams for the individual axles to form a general force diagram in which firstly all external forces acting on the vehicle combine with the frictional forces arising and with the flange pressure P to form a continuous closed force polygon, and secondly in the location diagram three resultants from these forces or remaining individual forces meet at one point, so that the sum of the moments of all forces acting on the vehicle relatively to a given point in the rail plane are equal to 0.

To fulfil the first condition the sum of all forces in the X, Y (Z) directions must be equal to 0, while fulfilment of the second condition calls for the sum of the moments to be equal to 0, as mentioned above.

In order to meet the last condition, the general force diagram must fulfil the conditions mentioned in chapter A :

for a four-wheeled vehicle,

$$L = \frac{l}{s}(H_2 \pm \zeta \cdot P_s) + \frac{Z}{2} - \frac{S}{2};$$

in general :

$$L = \Sigma\,\frac{li}{s} \cdot H_i \pm \frac{l}{s} \cdot \zeta \cdot P_s + \frac{Z}{2} - \frac{S}{2}.$$

It is often possible to ascertain or obtain realisation of the condition « Sum of all moments in the horizontal plane to equal zero », more easily from the *location diagram*, in which the sum of the moments of all forces occurring around anyone point, e. g. point 1, must be zero; also it must be proved for the striking axles, that the sum of the vertical forces equals 0; this being proved, according to chapter C, either graphically by means of

the R_a and R_n lines, or mathematically by the formula :

$$\overline{NU} + 0.145\,\overline{OF} + 0.43\,\overline{F\mathfrak{M}} = \frac{Q_1}{4}$$

(see page 787 and fig. 7*a*).

To ascertain the forces satisfying these three conditions, an arbitrary Δ R or H is assumed which in the first instance should approximately fulfil the simplest condition in the location diagram, viz. « Sum of the moments in the horizontal plane equal to zero »; in the case of geometrically-guided vehicles with given M line (vertical line of the centres of friction), one may start with the Δ R, otherwise it is better to start with the H of a non-striking axle. In the latter case, the H being selected, the corresponding centre of friction, the corresponding Δ R, and the line of the centres of friction are known, as outlined in chapter C, 2*a* and *b;* the condition « Sum of the moments in the horizontal plane = 0 » will determine the direction of fN'_1, the centre of friction M_I and therefore, all forces corresponding to the H or Δ R adopted. If the adopted Δ R or H does not fulfil the latter condition (sum of the vertical forces on the striking axle = 0), a more suitable Δ R or H should be adopted and the same construction carried out with it.

If this does not fulfil the aforesaid condition, the correct Δ R or H and the corresponding other forces should be determined by interpolation. How this method is to be applied will be shown below by means of individual examples of different types of vehicles and external forces.

1) Short four-wheeled vehicles.

(a) Vehicle with and without external transverse forces.

Example 1 :

Vehicle having a wheelbase of $l = 2.60$ m., wheel diameter = 1.20 m., i. e. $r = 0.60$ m.; R = 200 m.

The two axles are not coupled; there are no external forces; the centrifugal force is therefore supposed to be balanced exactly by the superelevation.

As the axles are not coupled, the centres of friction of axle I and of axle II do not coincide, but must be located vertically above each other on the same parallel to the axis. (See Chapter C, pp. 774 and 775).

General method of solving the problem.

For the solution of this problem, different values of H_2 are first assumed, and then the B-line for the force diagram of axle II is drawn up, as explained in chapter C, pp. 781-2, (only a short line is required). Then the $\frac{l}{s}$ line C_0G is drawn from C_0 (fig. 10*a*), with an inclination of cotan $\gamma = \frac{l}{s}$ on the longitudinal axis of the vehicle.

In order to have equilibrium in the horizontal plane we must have : $H_2 : L = s : l$, i. e. the parallel to the base C_0A through the end point B of the segment $\overline{AB}$ representing the magnitude H_2 must meet the $\frac{l}{s}$ line at G in such a manner that the perpendicular dropped from G on the base line parallel to the longitudinal axis of the vehicle cuts off the sum of the forces, L.

Empirically at least, two H_2 should be assumed, and as shown in figures 4, 4*a* and 7, 7*a*, force diagrams and location diagrams are drawn for the two axles of the vehicle and joined together. The H_2 adopted will generally not fulfil the conditions for the equilibrium in the vertical plane (fig. 7*a*).

The solution which fulfils also this condition and gives the correct H_2 will be obtained easily by interpolation.

The solution.

As a first empirical H_2, an H_2 of such a magnitude is assumed ($\overline{A_{5.45}\ B_{5.45}}$ in fig. 10*a*), that $B_{5.45}$ is the point of intersection of the B-line with the $\frac{l}{s}$ line drawn from C_0. For this purpose, the B-line or a portion of it is drawn, according to fig. 4*a*, and the $\frac{l}{s}$ line is set off from C_0 which cuts the B-line at $B_{5.45}$. $B_{5.45}$ is located 5.45 cm. above the horizontal through C_0 and gives, therefore, an $H_2 = \overline{A_{5.45}\ B_{5.45}}$ of 5.45 cm. = 0.136 Q = $H_{5.45}$. For this particular $H_2 = H_{5.45}$, according to the formula $L = H_2 \cdot \frac{l}{s}$ (page 778), $L_{5.45} = H_{5.45} \cdot \frac{l}{s} = \overline{C_0 A_{5.45}}$, which means that the end point of the force fN' is located at a distance equal to $\overline{C_0 A_{5.45}}$ from the C line, or, in other words, that $N_{a\,(5.45)}$ is vertically below $A_{5.45}$. The magnitude of the force $fN'_1 = \overline{A_{5.45}\ N_{a\,(5.45)}}$ is found to be $fN'_1 = f(Q - H_{(5.45)} \frac{r}{s})$, as the transverse force $H_2 = H_{5.45}$ is transmitted from the vehicle frame to the axle I. This corresponds with our values $f = \frac{1}{4}$, $r = 0.60$ of a segment $\overline{N_{a\,(5.45)}\ A_{5.45}}$ of $\left(10 - \frac{5.45}{10}\right) = 9.455$ cm.

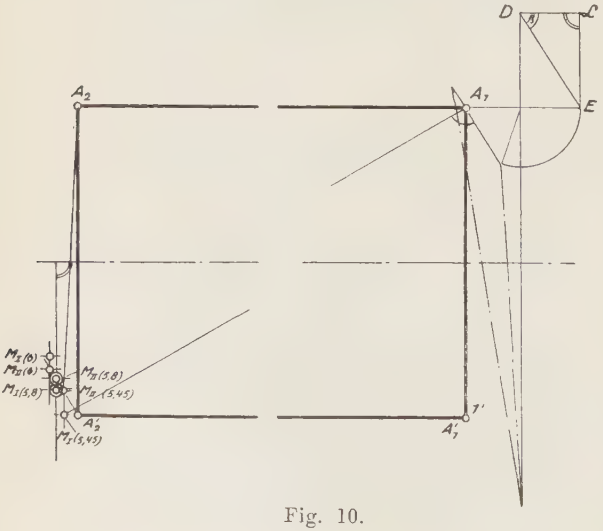


Fig. 10.

This gives us the direction and the end point $N_{a\,(5.45)}$ of fN'_1 in the force diagram. This fN'_1 for $H_2 = 5.45$ cm. = 0.136 Q meets the first condition, « the sum of the moments in the horizontal plane (location diagram fig. 10) = zero », the direction and the end-point $N_{a\,(5.45)}$ of fN'_1 being thus settled. The force fN_2 is represented by a segment of the same magnitude; fN'_2, however, becomes $10 + \frac{5.45}{10} = 10.545$ cm.

The centre of friction $M_{(5.45)}$ (see figure 10) of axle II is common to both force directions fN_2 and fN'_2.

The centre of friction for axle I must also be located on this main radius (i. e. on the perpendicular from the centre of the curve on the longitudinal axis of the vehicle) through this point, where $M_{I(5.45)}$ is the point of intersection of the horizontal through point 1' with the main radius mentioned above, as fN'_1 is perpendicular to the longitudinal axis of the vehicle.

If we draw now, as mentioned in chapter C, page 788, for these p and ΔR, the R_t and R_n line and set off the R_n line from $Q_{5.45}$, it will cut the line $\overline{A_{5.45}\ N_{5.45}}$ at point $N_{b\,(5.45)}$.

The following relation now holds for $H_2 = 0.136Q$ (= 5.45 cm.) : In order to have equilibrium in the horizontal plane, $N_{5.45}$ must be located at point $N_{a\,(5.45)}$ and to have equilibrium in the vertical

plane, $N_{5.45}$ must be located at point $N_{b\,(5.45)}$.

As the two points do not coincide, $H_2 = 0.136Q$ is not yet the correct H_2.

A suitable, and somewhat larger H_2, say $H_2 = 0.15\ Q = 6$ cm. is then assumed. At a distance $H_2 = 6$ cm. from the base line, we have, in the force diagram, the points B_6 and G_6, the latter as intersection of the $\frac{l}{s}$ line with the horizontal drawn through B_6. In order to have equilibrium in the horizontal plane, $N_{a(6)}$ must be located vertically below G_6 (as G_6 is located to the left of C_0 by the amount $L = H_2 \cdot \frac{l}{s}$ corresponding to $H_2 = 0.15\ Q = 6$ cm.) and on the circle drawn around A_6 with $fN'_1 = \left(10 - \frac{6}{10}\right) = 9.4$ cm. as radius.

The centre of friction $M_{II\,(6)}$ corresponds to the segment $H_2 = 0.15\ Q = 6$ cm. (see fig. 10), and the centre of friction $M_{I(6)}$ (in figure 10) corresponds to the direction $A_6N_{a(6)}$ just found.

The R_n direction to this $M_{I(6)}$ is found practically parallel to the preceding one, and is set off at Q_6. Q_6B_6 is now 10.6 cm. The new R_n line intersects the A_6N_{a6} line at $N_{b\,(6)}$. In order to have equilibrium in the horizontal plane with $H = 6$ cm. (0.15 Q), N_6 must be located at the point $N_{a\,(6)}$, and to have equilibrium in the vertical plane N_6 must be located at point $N_{b\,(6)}$. Consequently $H_2 = 0.15\ Q$ is already too great. We now join $N_{a\,(5.45)}$ to $N_{a\,(6)}$ and $N_{b\,(5.45)}$ to $N_{b\,(6)}$ and with the point of intersection N_r the correct point N is obtained.

From this point N_r a line is drawn vertically upwards as far as to intersect the $\frac{l}{s}$ line at G_r, and from G_r come back horizontally as far as to intersect with the

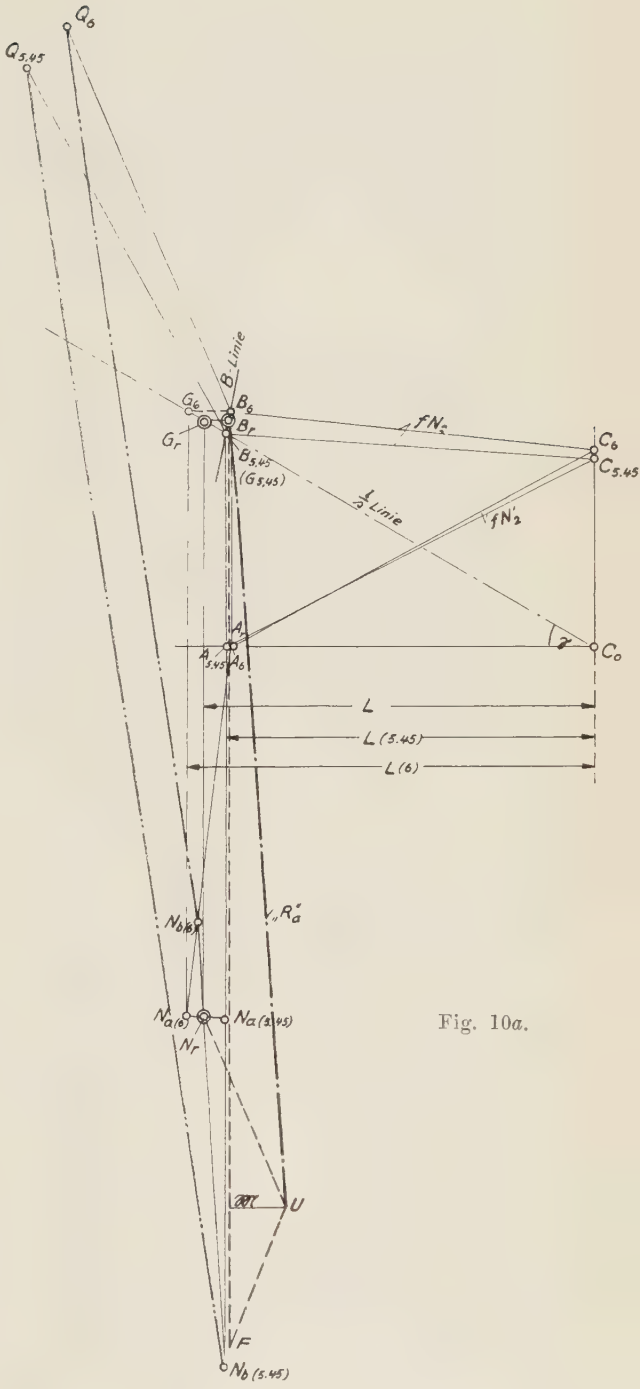


Fig. 10a.

B line at B_r, which gives $H_2 = 5.8$ cm. = 0.145 Q as the H_2 fulfilling the conditions of equilibrium in the vertical and horizontal planes. The corresponding centres of friction $M_{II\,(5.8)}$ for axle II, and $M_{I\,(5.8)}$ for axle I (see fig. 10) are then easily found, as outlined above. All necessary forces can now be deduced from the force diagram. The flange pressure $\mathfrak{N}$ at point $\mathfrak{A}$, which acts obliquely upwards at an angle of $\beta = 60^0$, is

$$\frac{P}{\sin \beta} = \overline{B_r F} \cdot \left(\frac{1}{\sin 60^0}\right) = \overline{B_r F} \cdot 1.16.$$

F is found according to figure 7*a* as shown dotted in figure 10*a*.

The description of the right construction of the force diagram appears tedious, but it is drawn up most rapidly and accurately.

Example 2.

If an external force, e. g. a transverse force P_s directed towards the inside of the curve, acts on the same vehicle above the rail plane it is necessary to consider that this force must be included in the force polygon of the location diagram, that the bearing pressures are altered, and that L becomes

$$L = (H_2 - \zeta P_s) \cdot \frac{l}{s}$$

The $\frac{l}{s}$ line therefore does not start from C_0, but from a point C_p located ζP_s above C_0 (see solution, fig. 11*a*).

(b) Short vehicle on which a tractive effort is exerted.

The tractive efforts which act on the frame above the rail plane also cause an uneven load of the wheels as they reduce the load on the leading wheels, and increase that on the trailing wheels. With long vehicles, and especially those with not very stiff springs, this effect is so

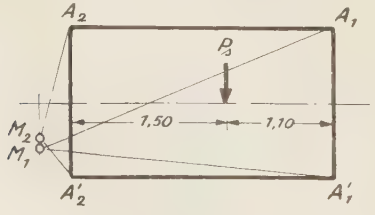


Fig. 11.

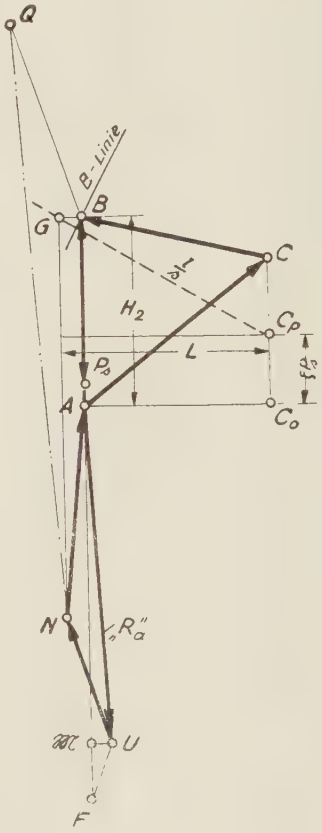


Fig. 11*a*.

small that it can be overlooked. With short vehicles and large tractive forces, however, it is important and must be taken into account.

In order to find at first whereto the centre of friction moves, when the tractive forces increase more and more, a large number of investigations were made based on the same simplified assumptions as when carrying out the consideration in the plane. Accordingly, the following preliminary examinations apply with the assumption of equal wheel loads and pressure-roller guidance; the axles are coupled.

With these assumptions the centre of friction for a vehicle of 2.60 m. wheel base, for $Z = 0$, is on the longitudinal axis of the vehicle 28 cm. or more exactly 27.5 cm. behind axle II (see fig. 1*a*).

Now, what is the magnitude of the tractive effort Z when the centre of friction is located on the line $M_{c'}$-$M_{c''}$ (fig. 12), say $\Delta R = 50$ cm. above the longitudinal axis of the vehicle ?

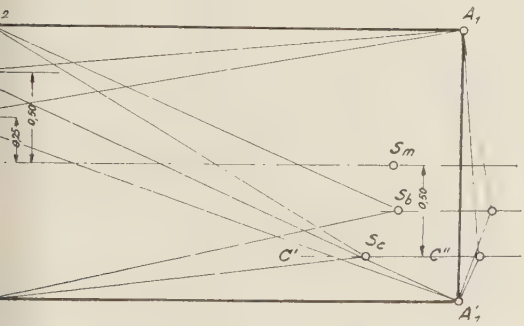


Fig. 12.

We know from chapter C 1 *a* that the point of intersection of the frictional forces fN_2 and fN'_2, and also the point of intersection of fN_1 and fN'_1 must then be on line c'-c'', 50 cm. below the longitudinal axis of the vehicle.

The centre of friction M_c must be fixed on line $M_{c'}$-$M_{c''}$ in such a manner that the frictional forces which it settles as to their direction, [all being assumed of equal magnitude (fQ)], are in equilibrium in the force diagram with a vertical directing force P and a horizontal tractive force Z, i. e. so that the condition tion

$$\frac{H_2}{L - Z/2} = \frac{s}{l}$$

is also fulfilled (fig. 12*a*).

Point F must be vertically above the point N of figure 12*a*.

When point M (see fig. 12) moves horizontally to the left by the small distance from $M_{c''}$ to $M_{c'}$, the part ANUG (see fig. 12*a*) of the force diagram will only vary very slightly, as the points A_1 and A'_1 are at a great distance from M_c and as the distance $M_{c''}$-$M_{c'}$ has hardly any influence in the directions of the forces fN_1 and

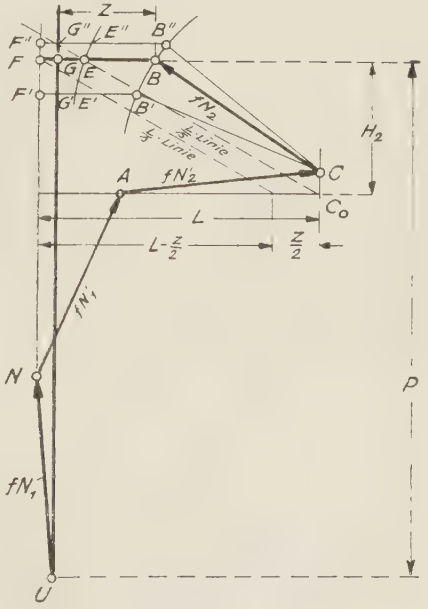


Fig. 12*a*.

fN'_1. Similarly, point A'_2 also is rather distant from M_c, so that the direction of fN'_2 is also varied very little and in any case remains very even. Starting with point A as a fixed point and as the origin of the force diagram, it, may, therefore, be assumed without too great an error that point C_0 of the force diagram (fig. 12*a*) retains its position whilst the centre of friction moves from $M_{c''}$ to $M_{c'}$. Point B, however, varies its position markedly, as it moves on the indicated arc of circle (which actually is only approximately an arc of circle) through B.

The directions of fN_2 corresponding to the two different positions of $M_{c'}$ and $M_{c''}$ (two are sufficient) are now set off in the force diagram and the segments $\overline{G'B'}$ and $\overline{G''B''}$ give the available tractive forces. Neither of the two solutions, however, fulfils the condition :

$$L = \frac{l}{s} \cdot H_2 + Z/2 \quad \text{or} \quad \frac{H_2}{L - Z/2} = \frac{s}{l},$$

the condition for equilibrium in the horizontal plane. The correct solution for equilibrium is found by means of the « E line », which is obtained by setting off to the right, starting from F' and F'', half the length of the corresponding Z, and by joining together the two points E' and E'' so obtained. The point of intersection E of the E line with the $\frac{l}{s}$ line drawn through C_0 is the final E; for now we have

$$\frac{H_2}{L - Z/2} = \frac{s}{l}.$$

The horizontal through E intersects the B line at point B, the direction of fN_2 is fixed by BC, and the centre of friction M_c is determined by this direction, and the direction of fN'_2 Z_c, with our assumed numerical values, will be found equal to 0.12 Q. M_c lies in fact on the parabolic arc passing through the points M_m, A_2, and A'_2.

If the centre of friction is required to be 0.25 cm. above the longitudinal axis of the vehicle, then the centre of friction is M_b and the force $Z_b = 0.05$ Q. If the force Z is increased, say to 0.16 Q, then the centre of friction will have reached point A_2.

Where does it, however, move if Z increases further ? It might be thought that it would move along its parabolic arc. This is, however, not the case, and is also impossible, because it must be considered that the direction of force fN_2 reverses at once with the change of the centre of rotation from one side of the axle II to the other, i. e., so that $-K_2$ changes immediately into $+K_2$ and similarly $-G_2$ into $+G_2$.

There occurs here an important change in direction of the curve of the centres of friction. For a large enough range of the tractive forces Z, the centre of friction is fixed at point A_2. Figure 13 shows how it moves still further for a very large Z ($Z > \frac{1}{7}$ of the weight of the vehicle).

As long as the centre of friction remains fixed at A_2, this condition means that the outside forces are not able to overcome the frictional force fN_2 : the frictional force fN_2 is not used up. As long as the centre of rotation is fixed at A_2, the construction of the force diagram is quite simple.

The directions of the frictional forces fN'_2, fN'_1, and fN_1 are fixed as long as the centre of friction remains A_2. They are simply added together. The force Z (represented in figure 13*a* for Z_d = 0.2 Q and in figure 13*b* for $Z_e = 0.5$ Q) is then added horizontally at the end point, the resultants are drawn in the lo-

cation diagram, figure 13, as far as the point of intersection W_d (for $Z_d = 0.2\ Q$) and W_e (for $Z_e = 0.3\ Q$). The connecting lines $W_d\ A_2$ and $W_e\ A_2$ give the direction of the forces $\kappa f N_2$ which closes the force polygon.

If the force Z exceeds 0.62 Q, it will then be of sufficient value to again overcome at point A the adhesive frictional force which rotated nearly by 150°; the centre of friction moves upwards but remains very closely to the left of axle II, because fN_2 becomes flatter and flatter.

If, for example, we adopt point M_p as centre of friction and combine the frictional forces fN_2 and fN_1 in the force diagram, figure 13*c*, and then the forces fN'_2 and fN'_1 and the resultant of the latter with Z, which is obtained from the force diagram, the final resultants must meet on the prolonged axle I, in the direction of the directional force (point W_p in figure 13).

As Z increases, the centre of friction will therefore move as is shown by the curve in figure 13. This curve will in principle retain the same shape when dealing with the problem in space, i. e., with two-point contact and when the alteration of the bearing pressures is taken into account. The curve is the locus for the centre of friction.

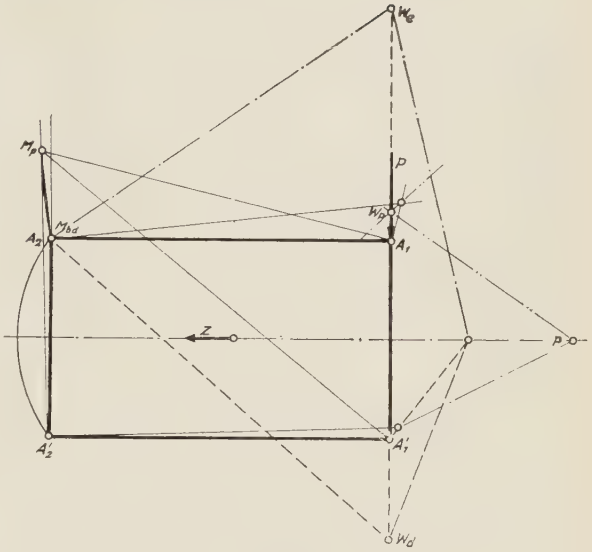


Fig. 13.

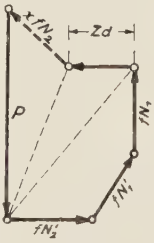


Fig. 13*a*.

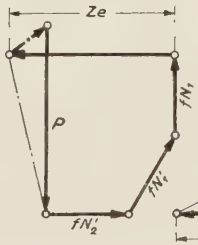


Fig. 13*b*.

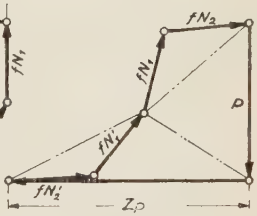


Fig. 13*c*.

Example :

As an example of the graphical solution for a short vehicle submitted to a tractive effort, we shall again assume the case of the vehicle given in example 1, to which a small tractive effort Z = 0.1 Q is applied (fig. 14). The axles are assumed to be coupled. Simplifications are no longer made, but all influences are considered. In the first instance, the effect of the tractive effort alters the wheel pressures, i. e. each wheel of the trailing pair receives an increase in load :

$$\Delta Q_z = \sim \frac{Z\,h}{2\,l} = \frac{0.1 \cdot 1.0}{2 \cdot 2.6} = 0.0194\ Q,$$

wherin h means the height of the drawbar above the rail head. The load on each front wheel will be decreased by the same amount.

It can be admitted, based on the above rough considerations, according to which (for Z = 0.16 Q) the centre of friction is located at point 2, that the centre of

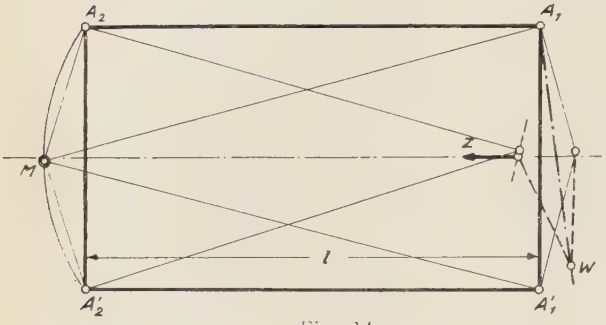


Fig. 14.

friction M will be on the arc between A_2 and A'_2 (see fig. 12). The value of H_2 should be chosen so that the $\frac{l}{s}$ line starting from the upper end point A_0 (fig. 14*a*) will pass between the points C_u and

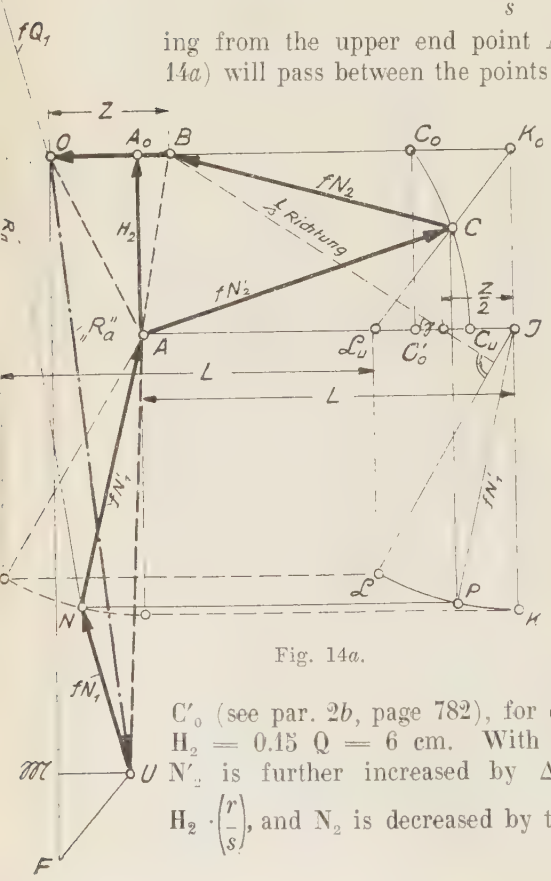


Fig. 14*a*.

C'_0 (see par. 2*b*, page 782), for example, $H_2 = 0.15\ Q = 6$ cm. With this H_2, N'_2 is further increased by $\Delta Q_h = H_2 \cdot \left(\frac{r}{s}\right)$, and N_2 is decreased by the same amount. If we now draw a circle around A as centre with fN'_2 as radius, the point C we seek will be on this circle. To point C_u corresponds an exactly horizontal force fN'_2, and therefore a centre of friction at A_2. To point C_0 corresponds an exactly horizontal force fN_2 and, therefore, a centre of friction at A'_2. Hence when C moves from C_u to C_0, the centre of friction of the axle II moves on a flat arc from point A_2 to A'_2. As, however, the centre of friction of axle I must always be the same as that of axle II, the direction of the frictional force fN'_1 will change evenly from JL (see fig. 14*a*) as far as the vertical JK, with JL perpendicular to $A_2A'_1$ in the location diagram (fig. 14*a*). Only one of the corresponding pairs of directional forces AC and JP fulfils the condition of equilibrium in the horizontal plane. This pair is found as follows :

The force fN'_1 in figure 14*a* being displaced parallel to the right from A to J, is able to rotate around J. J has been displaced parallel from A to the right by the distance $L = \left(\frac{l}{s} \cdot H_2 + \frac{Z}{2}\right)$. When added together, the two horizontal components of fN'_2 and fN'_1 must be equal to this quantity L. As when the end points C of the force fN'_2 moves from C_u to C_0, the end point P of the force fN'_1 moves at the same time from L to K, the corresponding position in which C and P are placed vertically above each other must be determined. For this purpose, the approximately straight motion LK is replaced by the exact straight motion L_u K_0 (L_u vertically above L on a level with AC_u, K_0 vertically above K, and J on a level with A_0BC_0). The point of intersection of the line C_uC_0 with L_uK_0 gives the point C, as well as, vertically

below it, the point P which correlatively fixes the direction for fN'_2 and fN'_1 in such a manner that for our adopted H_2, equilibrium in the horizontal plane is ensured. The force $\overline{JP} = fN'_1$ we have thus ascertained is set off in value and direction from point A as far as N. The three directions so obtained of the forces fN'_2, fN'_1 and fN_2 give, for our adopted H_2, the centre of friction corresponding to the equilibrium in the horizontal plane.

If the R_n direction is now drawn up, according to figure 7*a*, and set off from point Q, it must pass through point N, otherwise an incorrect H_2 has been adopted. The same procedure will then have to be carried out again with a slightly larger or smaller H_2 (say $H_2 = 0.14$ Q). The correct H_2 of the example, i. e. 0.146 Q is finally obtained by interpolation. The centre of friction is very nearly located on the longitudinal axis of the vehicle, whereas without tractive effort, it was 44 cm, beneath it. In other words, it has moved upwards.

If the calculation is to be carried out with a medium-sized Z (say Z = 0.2 to 0.75 Q *with* consideration of the alteration of the wheel pressures due to Z and H), the centre of friction is fixed at point A_2, and all the force directions are given immediately. The force diagram can be drawn up immediately. Even if there should arise transverse forces P_s in addition, the centre of friction will remain at point A_2.

The figures 15 and 15*a* will be obtained if Z is very large (Z > 0.75 Q) say Z = 0.85 Q. M proceeds rapidly outwards, H_2 becomes smaller, so that it is possible to overlook its influence on the fN forces.

There are also the cases of individual motor-driven axles (electric locomotives).

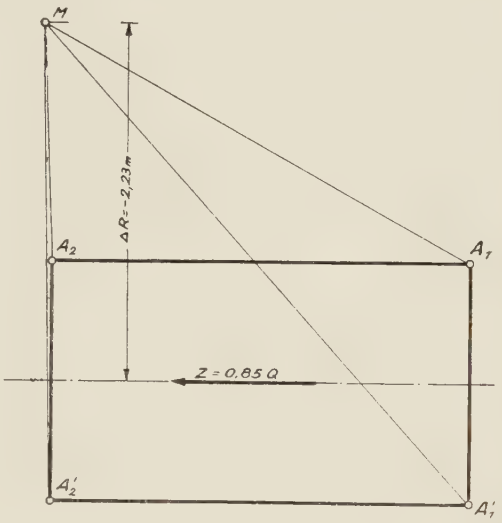


Fig. 15.

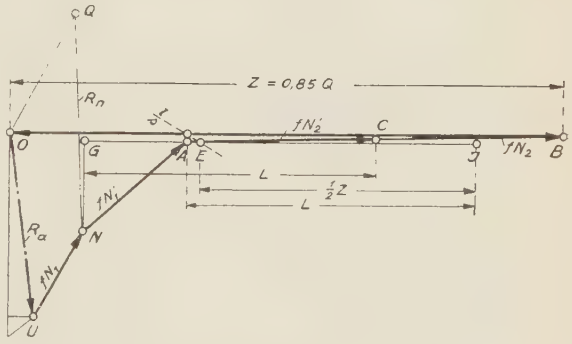
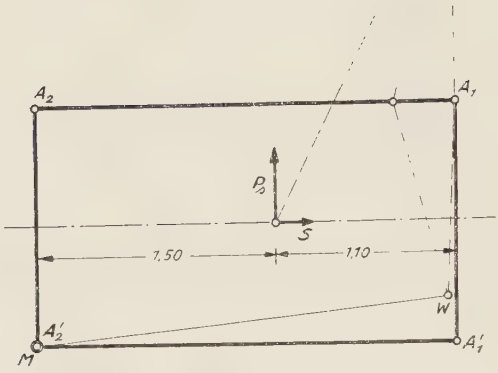


Fig. 15*a*.

In this case the Z force of a driving axle must only be absorbed by that axle, and consequently this Z must only be included in the force-diagram of the driven axle. A corresponding solution is found in the case of braking forces, as braking forces can be considered as negative tractive forces. Already with small braking forces (S > 0.05 Q) and up to rather large (S < 0.55 Q) braking forces, the centre of friction is fixed at point A_2,

Fig. 16.



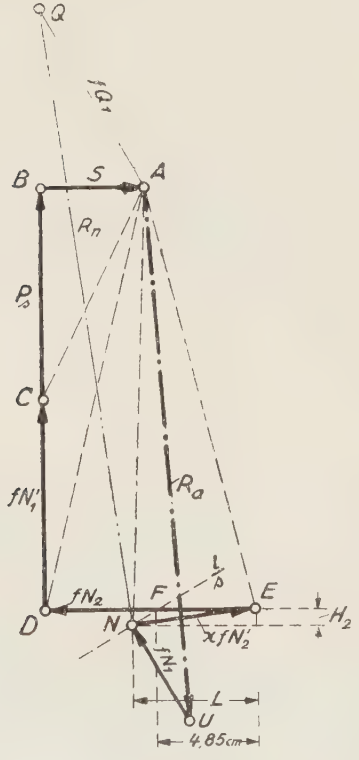


Fig. 16*a*.

and only for strong braking forces, (> 0.55 Q) will it drop below point A'_2 towards the inside of the curve.

Example :

Take the same vehicle, with coupled axle braking force $S = 0.125$ Q, transverse forces $P_s = 0.25$ Q (fig. 16).

In this case — $H_{II} = H_2$ becomes very small, as the share of P_s falling to axle II opposes the force ($-H_{II}$) and reaches nearly the same value; it is therefore possible to overlook the influence of H_2 on the alterations of the wheel loads Q_2 and Q'_2. However, account must be taken of the alteration of wheel loads caused by P_s.

As the centre of friction M is fixed at point A'_2, the forces S, P_s, fN'_1 and fN_2, of known value and direction, can be set off in succession in the force diagram, figure 16*a*, giving the figure ABCDE. From A the direction of fN'_1 is then drawn, fQ_1 is set off on it as far as Q, from which point the R_n line is drawn. On this line must be located point N (1st locus). As in this case N is the end point of the force of adhesive friction $\kappa fN'_2$, the second locus is easily found from the known equation :

$$L = \left(\frac{l}{s}\right)(H_2 + \zeta \cdot P_s) - \frac{S}{2}$$

(condition of the sum of the moments = 0; see page 778, chapter C 1).

In our case :

$$L = \left(\frac{l}{s}\right) \cdot H_2 + 4.85 \text{ cm.}$$

$$-\frac{S}{2} + \left(\frac{l}{s}\right) \cdot \zeta \cdot P_s = 4.85 \text{ cm.}$$

is set off horizontally from E on the other side as far as F, and the $\frac{l}{s}$ line is

produced up to its intersection with the R_n line, at N. $\overline{NE}$ will then give the value and direction of the adhesive friction at A'_2, and $\overline{NU}$ the value fN_1. U is the point of intersection of the R_a line with the fN_1 direction. The solution is correct, for all forces combined in the location diagram meet at one point W.

The short vehicle with several pairs of wheels.

Vehicles with several pairs of wheels can also be « statically » guided, i. e. only by the striking leading wheel. In most cases, however, vehicles with several pairs of wheels will be of such a length as to be geometrically guided, i. e. the leading pair will strike to the outside, one rear-axle to the inside, especially if the narrow gauge has been laid according to the new track specifications. It might just be a case of a six-wheeled vehicle with short wheelbase in a rather wide gauge, and we will now carry out the calculation graphically.

The wheelbase of the six-wheeled vehicle is assumed to be 4.50 m., having a centre axle with side play, the gauge of the track to be wide enough to enable free running of the trailing pair of wheels.

It must first be examined whether the side play of the middle pair of wheels is such that the latter will strike the outside rail, or whether it is guided by the frame. In the first case the axle will only cause a definite moment $\mathfrak{M}_{II}$ for the equilibrium condition in the horizontal plane, and no horizontal transverse force H_{II} on the axle guard : in the case of the non-striking axle it causes a definite $— H_{II} = H_2$ and $\mathfrak{M}_{II}$ (only slightly depending on p_2).

(a) The middle axle has sufficient side play to strike to the outside.

The middle axle shall be considered by itself. In the horizontal plane at rail level the two frictional forces fN_2 and fN'_2 must form, with the forces at the flange pressure point $\mathfrak{A}_2$, a closed force polygon in the force diagram, as there are no outside forces and no H_{II} to be overcome. Thus the force triangle ABC (fig. 17) for axle II will be formed,

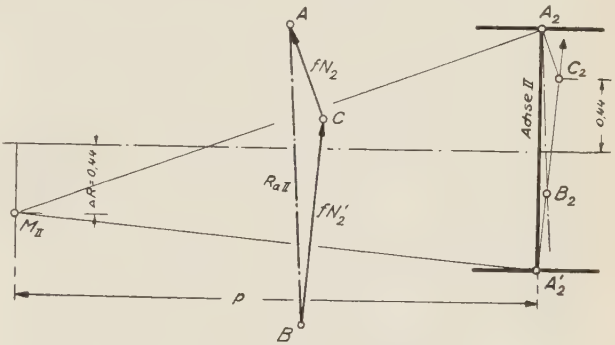


Fig. 17.

wherein, as before, the force AB is the resultant R_a of the horizontal components of the forces acting at the flange pressure point $\mathfrak{A}_2$. H_{II} is equal to 0, as axle II does not press on any side against the axle guards. As the position of M_{II} is still unknown, neither the direction nor the position of AB are known. In the triangle ABC, only the value of $\overline{BC} = fN'_2 = fQ_2$ is known. We know in addition that, according to page 787 it is necessary that $\overline{AC} + 0.235\ \overline{AB} = fQ_2 = \overline{BC}$. A triangle similar to this triangle ABC is built in the location diagram and forms figure $A_2B_2C_2$. It will be found by trial and error that for R = 200 m., r = 0.6 m., and for cases of p = 2 to 8 m. (which most usually arise) a striking axle (without side forces and without H-forces at the axle) is in equilibrium when the centre of friction M_{II} has been

assumed 0.44 m. below the longitudinal axis of the vehicle, i. e. when $\Delta R_{II} = +0.44$ m. This value is only valid for $r = 0.60$ m.; for $r = 0.50$ m., for instance, it increases already to $\Delta R = +0.52$ m. The ΔR varies if the wheel pressures at the axles have different values, or if the axle has to take up the H, P_s, Z, or S forces. Consequently a horizontal line drawn

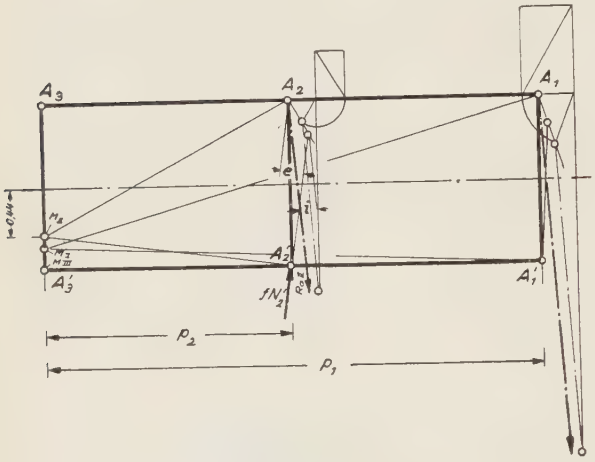


Fig. 18.

0.44 m. below the longitudinal axis of the vehicle is a locus for M_{II}, and therefore the position of the centres of friction in the direction of the longitudinal centre line of the vehicle, i. e. the position of the main radius depends on the axle III. It was previously held that it coincided with axle III; recently BÄSELER and BECKER have proved that it is not on the axle itself, but very near it. That is correct. In order to ascertain the location of M_{III}, it is here assumed that the perpendicular on the longitudinal axis of the vehicle coincides with axle III, i. e. that its distance from the front axle $p_1 = 4.50$ m. The moment $\mathfrak{M}_{II}$, which moreover varies very little when M_{II} comes a little behind or ahead of the axle III, then becomes $\mathfrak{M}_{II} = fN'_2 \cdot e = R_{aII} \cdot i$ (see fig. 18) $= 0.052$ Qm, as $e = 0.208$ m. in figure 18, and since we assume that $f = \frac{1}{4}$ and $N'_2 = Q_2 = 1$).

As p_1 is twice the value of p_2, $\mathfrak{M}_I$ must be about half the value of $\mathfrak{M}_{II}$; it will become even a little smaller as it is also influenced by a negative H_{III} = a positive H_3 :

$\mathfrak{M}_I =$ approx. 0.02 Qm (the final solution gives $\mathfrak{M}_I = 0.022$ Qm).

$\mathfrak{M}_{III} = fN_3 \cdot s =$ approx. 0.375 Qm; $\mathfrak{M}_{II} =$ approx. 0.052 as above.

$H_3 \cdot l = \mathfrak{M}_I + \mathfrak{M}_{II} + \mathfrak{M}_{III} =$ approx. 0.447 Qm.

$$H_3 = \frac{0.447}{l} = \frac{0.447}{4.5} = 0.1\ Q \text{ approx.}$$

Now, for this small H_3, which has been ascertained approximately, the centre of friction for axle III, will, however, be already fixed at point A'_3 as shown by the following consideration : If we draw the force triangle for axle III, by setting off $H_3 = +0.1\ Q = 4$ cm., and the circles with fN_3 and fN'_3 radii, a right-angled triangle is just obtained, for under the influence of H_3, fN_3 becomes

$$fN_3 = f\,[\,Q - (r/s)\,H_3\,] = \frac{1}{4} \cdot \left(Q - \frac{0.6}{1.5}\,H_3\right) = \frac{1}{4} \cdot [1 - (2/5) \cdot 0.1]\,Q$$
$$= 9.6 \text{ cm.}; \quad fN'_3 = 10.4 \text{ cm.}$$

Now, $9.6^2 + 4^2 = 10.4^2$; therefore the triangle in question is a right-angled triangle. Consequently fN_3 is horizontal in direction, so that the centre of friction must be at point A'_3. To the forces triangle AED (fig. 18*a*) is then added the for-

ces triangle ABC for axle II whose centre of friction lies at point M_{II}, 44 cm. below the longitudinal axis of the vehicle. This gives for axle I the equation :

$$\mathfrak{M}_I + \mathfrak{M}_{II} + \mathfrak{M}_{III} = L \cdot 1.5 = - H_{III} \cdot 4.5 = H_3 \cdot 4.5.$$

$L = 3 \cdot H_3$, L meaning this time the sum of the three distances $\overline{ED} + \overline{CF} + \overline{NO}$ (fig. 18*a*). As according to the diagram, $\overline{ED} = 9.6$ cm. and $\overline{CF} = 1.5$ cm., there would remain for $\overline{NO}$, with $H_3 = 0.100\ Q = 4.0$ cm., $\overline{NO} = 3 \cdot 40 - 9.6 - 1.5 = 0.9$ cm., i. e. N would be in the force diagram 9 mm. to the left of the vertical DA. Thus the direction of AN would be obtained, as well as the centre of friction of the axle I at point M_I, and the state of equilibrium in the horizontal plane corresponding to the approximately found $H_3 = 0.1\ Q$ would be drawn.

In order to ascertain whether in this state there is also equilibrium in the vertical plane, the R line for axle I would have to be drawn. It passes very close to point N. A very small correction showed that H_3 was 0.095 Q instead of 0.100 Q, and that $\overline{NO} = 0.3$ cm. (see table No. 2).

This small difference does not alter the force and location diagrams in practice.

(b) The middle axle does not strike (figs. 19 and 19*a*).

It has been shown in Chapter C 2*a*, for the case of the non-striking axle free from influence, along which line the centre of friction M is displaced when H varies (M line in fig. 4). This line is set off for the axles II and III in the location diagram. For axle III two branches of the M line are drawn, the one for a positive, and the other for a negative H_3. If the main radius on which all centres of friction are located were now known, everything would be known. To get an idea as to where the main radius is to be found, the force relations should be considered according to the condition : Sum of all force moments about point I equal to 0. If the M vertical (main radius) lies closely to the rear of axle III or on axle III, the condition of equilibrium : $-\mathfrak{M}_I - H_3 \cdot g_3 + H_2 \cdot g_2 = 0$ cannot be fulfilled in this case (g stands for the distance of the resultant H_3 or H_2 from axle I). As a matter of fact, $\mathfrak{M}_I$ is already known to be very small if no external forces are acting, and is for $p =$ 4.5 m. say 0.02 Qm. $H_2 \cdot g_2$ remains nearly constant and equal to $0.47 \cdot 2.0 =$ 0.94 Qm.

$H_3 \cdot g_3$ should consequently become approx. $-0.94 + 0.02 = -0.92\ Qm$. If, however the M vertical proceeds from the left to the right until it coincides with axle III, the value ($-g_3$ increases into infinite, whereas the value H_3 decreases down to 0, and the value $H_3 \cdot (-g_3)$ increases permanently up to the value

$$\lim_{g_3 = \infty} [H_3 \cdot (-g_3)] = -(0.25 \cdot 1.5)\ Qm = -0.375\ Qm.$$

This value is, however, far too small when compared with the $H_3 \cdot g_3 = -0.92\ Qm$ required. If the main radius proceeds to the right side of axle III, then H_3 will change its direction, but g_3 also changes its position with respect to axle I, going from the left to the right; the product $H_3 \cdot g_3$ consequently remains negative and increases rapidly at the same time as $-H_3$.

For the M vertical drawn in the location diagram, which in the 1 : 10 location diagram is only 1 cm. to the right of axle L, the condition $-H_3 \cdot g_3 + H_2 \cdot g_2 - \mathfrak{M}_I = 0$ is fulfilled. The points of intersection with the M lines are denoted by M_{III} and M_{II}.

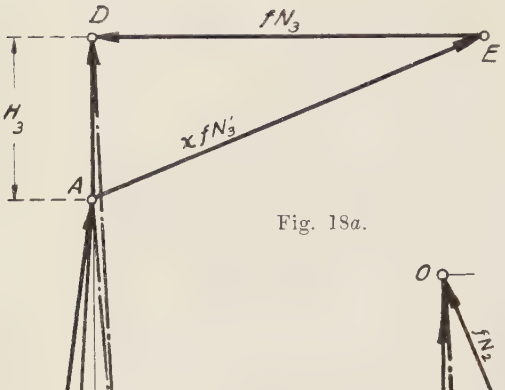


Fig. 18a.

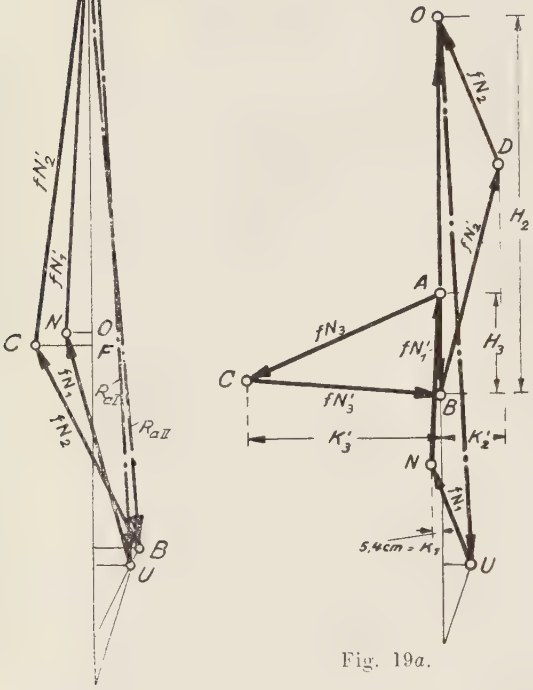


Fig. 19a.

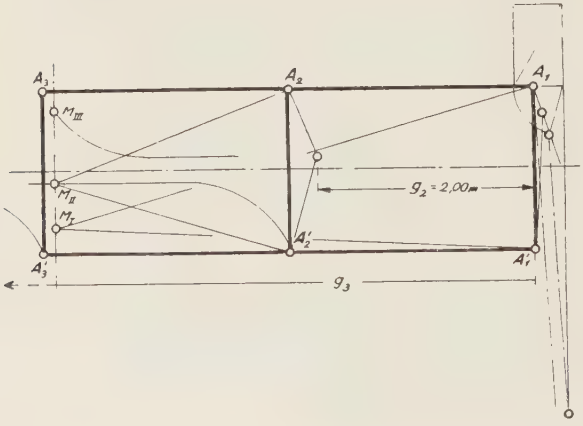


Fig. 19.

Fig. 20a.

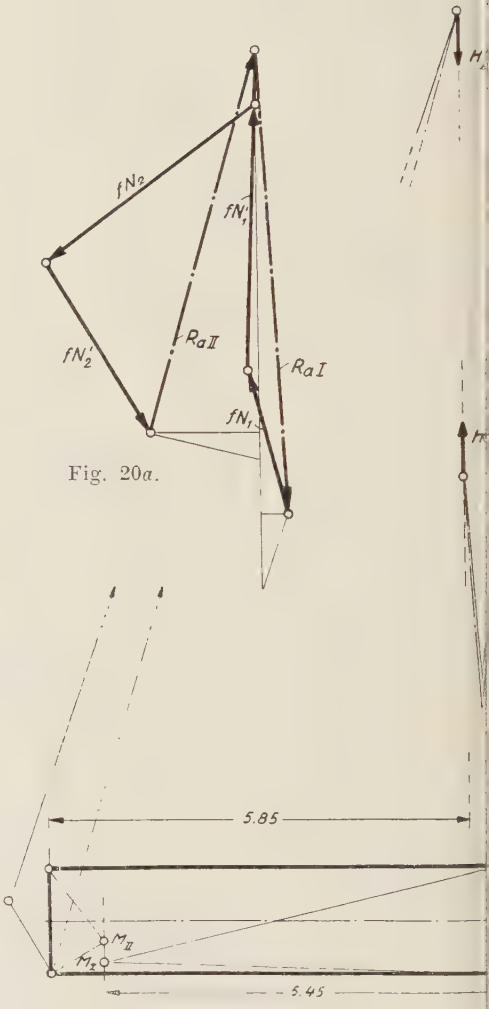


Fig. 20.

It is now easy to draw the force diagram with these centres of friction, and we get :

$-H_3 = -5.12$ cm., $+H_2 = 18.6$ cm.

Similarly, the X-components of the forces fN'_3 and fN'_2, which we shall call K'_3 and K'_2, can also be read off the force diagram. We have therefore again the following equation

$$L \cdot s = (K'_1 + K'_2 + K'_3) \cdot 1.5 = H_3 \cdot l + H_2 \cdot \left(\frac{l}{2}\right).$$

$$K'_1 = \frac{4.5}{1.5}\left(H_3 + \frac{H_2}{2}\right) - (K'_2 + K'_3).$$

$$K'_1 = 3 \cdot (-5.12 + 9.3) - (9.5 + 3.10).$$

$$K'_1 = 0.54 \text{ cm.}$$

In the force diagram, therefore, the point N is 5.4 mm. to the left of the vertical line OAB. The inclination AN of the frictional force fN'_1 gives in the location diagram a $\Delta R_I = 0.52$ m. for the centre of friction M. For this position of M_I, the R_n line passes also through point N.

3) The long vehicle in the 200-m. curve.

Having so far discussed the short vehicle, we will now consider the behaviour of a long vehicle on a sharp curve. The calculations and graphic representations are much simpler because the distance of the centre of friction from the first striking axle is given geometrically. We will consider a vehicle having two axles with side play, and a wheelbase of 6.20 m. (see fig. 20); proceeding as on page 776, it is found by calculation that p is 5.45 m. for R = 200 m. and σ (including the lateral displacement of the axles) = 73 mm. The diameter of the wheels of this vehicle is 1.00 m. instead of 1.20 m. as heretofore. The parallel to the axle direction, drawn at a distance $p = 5.45$ m. from axle I, is, therefore, a first locus for the centres of friction (fig. 20). The fact that we know this locus from the beginning greatly facilitates the solution. We now only require to locate the centres of friction on this locus in such a manner that the two conditions for the equilibrium of the vehicle :

$H_I = -H_{II}$ and $\mathfrak{M}_I + \mathfrak{M}_{II} = H_{II} \cdot l$ are fulfilled.

The determination of the position of M_I and M_{II}, i. e. the determination of the quantities ΔR_I and ΔR_{II} is best done graphically. According to chapter C and figure 9*a*, we ascertain for different ΔR_I, the H_I and $\mathfrak{M}_I$ required for the equilibrium of axle I, and H_I is plotted in terms of ΔR (fig. 22). In the same manner the H_{II} required for equilibrium at axle II are ascertained for different ΔR_{II} and plotted in the same figure (No. 22). In another diagram (figure 21) the sum of the moments $(\mathfrak{M}_I + \mathfrak{M}_{II})$ is plotted in terms of H, and the $(\mathfrak{M}_I + \mathfrak{M}_{II})$ line is made to intersect the $H \cdot l$ moment line The point of intersection gives the required $H = H_I = -H_{II}$. The values of ΔR_I and ΔR_{II} are read off figure 22, corresponding to the ascertained value of H.

Detailed method.

According to chapter B and figure 9*a*, the H for the axle I corresponding to the

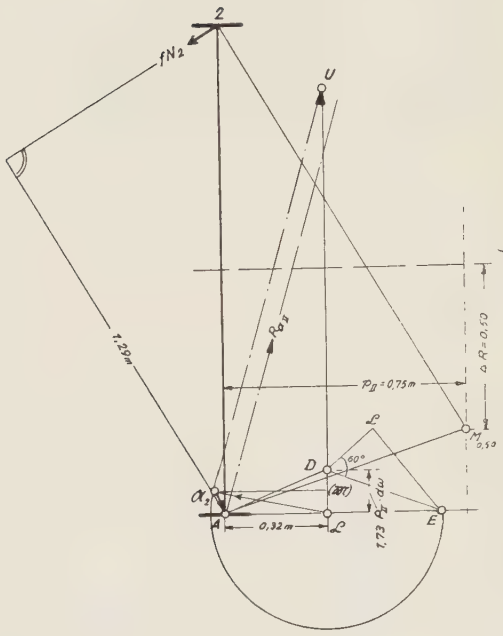


Fig. 20*b*.

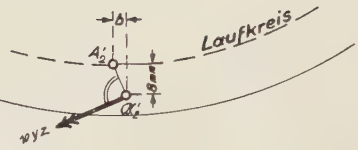


Fig. 20*c*.

Fig. 20*d*.

two $\Delta R_I = 0.75$ and 0.40 m. was found to be $H_I = 0.228$ Q and 0.131 Q, and the corresponding $\mathfrak{M}$. $\mathfrak{M}_I = 0$ and $\mathfrak{M}_I = 0.0209$ Q*m*.

For axle II, the H_{II} and $\mathfrak{M}_{II}$ required for equilibrium are drawn for $\Delta R_{II} = 0.50$ m. and 0.25 m. in a manner quite similar to that outlined in chapter C and figure 9*a*.

As unlike the construction given in chapter C, the centre of friction in this case is to the right of, and very close to, the axis, the construction for $\Delta R_{II} = 0.50$ m. will be carried out again, making use of figures 20*b*, 20*c* and 20*d* :

When the vehicle describes an angle $d\omega = \frac{1}{100}$, the rail lateral pressure point moves in the horizontal plane from A to $\mathfrak{A}_2$ (see the location plan, fig. 20*b*, where $A\mathfrak{A}_2$ is perpendicular to $AM_{0.50}$), and in the vertical plane from D to A where DA is parallel to the frictional force drawn in figure 20*c* at the diagramatically shown wheel-flange, so that $\overline{AC} = 8 \cdot d\omega \cdot \frac{R}{r} = 8 \cdot \frac{1}{100} \cdot \frac{200}{0.50} = 32$ mm. and $\overline{DC} = \tan \beta \cdot p_{II} \cdot d\omega$. In the horizontal projection, the rail pressure point moves from C to $\mathfrak{A}_2$. If we make $\overline{CU} = 4\ \overline{EL}$ (see page 785), then $\mathfrak{A}_2U$ becomes the di-

rection of R_{aII}. The direction of R_{nII} is found by the general formula given on page 787 : $0.145\,\overline{OF} + 0.43\,\overline{F\mathfrak{M}} = \xi \cdot \overline{OU}$ · ($\overline{O_hF}$, $\overline{F\mathfrak{M}}$ and O_hU_h are given by figure 20*d*, similarly $0.145\,\overline{CU} + 0.43\,\overline{C(\mathfrak{M})} = \xi\,\overline{\mathfrak{A}_2U}$. ($\overline{CU}$, $\overline{C(\mathfrak{M})}$ and $\overline{\mathfrak{A}_2U}$ are given by the location diagram, figure 20*b*).

From this ξ is obtained as 0.172.

The direction of the R_{nII} line is found by setting off distances at a scale of 0.172 : 1, from one point (e. g. from O_h) in the direction of the force fN'_2, i. e. in the direction O_hQ_h, and in the direction of R_{aII}, i. e. in the direction O_hU_h, and joining the end points, as shown in figure 20*d*. In figure 20*d*, the construction with this R_{nII}-direction for ascertaining the required H_{II} has been drawn in the exactly same manner as in figure 9*a*. $\overline{O_0N_0}$ = 10 cm, through N_0, the R_h line,

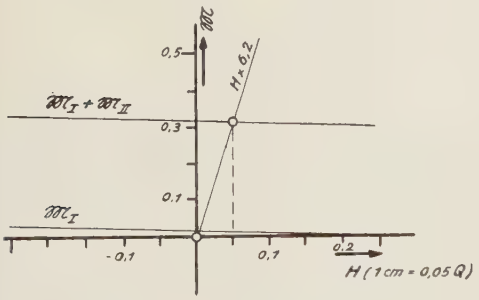


Fig. 21.

is drawn as far as W_h, from W_h parallel to $\overline{O_0N_0}$ as far as Q_h, the parallel to fN'_2 through Q_h cuts off on the vertical through O_0 the distance $O_h\,O_0 = H_{II} = -0.228$ Q. The force fN_2 becomes $= f(Q + H_{II} \cdot \frac{r}{s} = \frac{1}{4} \cdot (1 - 0.228\,\frac{0.50}{1.50}) = \frac{1}{4} \cdot 0.924 = 0.231$ Q and, in the location diagram, figure 20*b*, fN_2 rotates wish a static moment $\mathfrak{M}_{II} = -0.231 \cdot 129 = -0.299$ Q*m*. If ΔR_{II} is taken as = 0.25 m., then the corresponding $H_{II} = -0.0188$ Q and $\mathfrak{M}_{II} = -0.3$ Q*m* are ascertained in the same manner.

In figure No. 22, the two ascertained H_{II} and H_I are plotted above the corresponding Δ R and connected to form the two H lines, i. e. the H_I line and the H_{II} line, which run nearly straight.

On the other hand, by means of the ascertained $\mathfrak{M}_I$ and $\mathfrak{M}_{II}$, the ($\mathfrak{M}_I + \mathfrak{M}_{II}$) are plotted in figure 21, in terms of H, and this line is produced until it intersects with the H · l = H · 6.2 = moment line. The point of intersection is at a distance H = 0.051 Q. The horizontal drawn at level H = 0.051 Q cuts off, in figure 22, $\Delta R_I = +0.574$ and $\Delta R_{II} =$

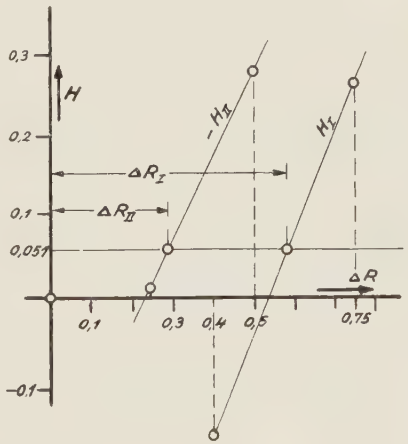


Fig. 22.

+0.288 m. Thus the correct centres of friction and the right H, and therefore all other forces, are found and combined in the force diagram 20*a*.

To check the results obtained the force directions and force values now given are drawn in the location diagram. The force diagram is closed, and in the location dia-

gram H_I and H_{II} meet and cancel each other out (fig. 20).

If any other transverse or tractive forces or both act on the long vehicle, these need only be added to the moment $H \cdot l$ of the moment line, and the corresponding final H is then cut off in the moment diagram.

If the vehicle is provided with middle axles the M and S line corresponding to the non-striking middle axle *(i)* gives immediately the centre of friction M_i, the value of H_i, and the value of the moment $\mathfrak{M}_i$; if the middle axle *(k)* has side play and guides itself in the transverse direction along the track, its centre of friction is again 0.52 m. below the axis of the vehicle, so that $\Delta R_k = 0.52$ m. and

$$\mathfrak{M}_k = fQ'_k \cdot \frac{0.23 \cdot 1.50}{p_k}.$$

It is, therefore, only necessary to increase the $\mathfrak{M}$ line by exactly given distances into an $(\mathfrak{M}_I + \mathfrak{M}_{II} + \text{etc.})$ line, and in the case of non-striking axles, to replace the $H \cdot l$ line by the $\Sigma (H \cdot l)$ line whilst all preceding graphical results for axle I and the striking rear axle remain unaltered. It is clear, therefore, that the additional axles do not cause any difficulties in the case of geometrically (front and rear) guided vehicles.

4) The statically guided vehicle, (i. e. only with the front axle) in the check rail track.

The forward displacement of the pressure-point of the striking wheel flange on the check rail is very much increased. The level of the check rail varies considerably for the different types of track. With type 6 on steel, the top edge of the check rail projects 11 to 12 mm. above the running rail level, with type 6 on wood the top edge of the check rail is at rail level, and for all other types it is almost flush with the running rail. Figure 23*b* shows the check-rail edge, the running rail and the lower part of the wheel.

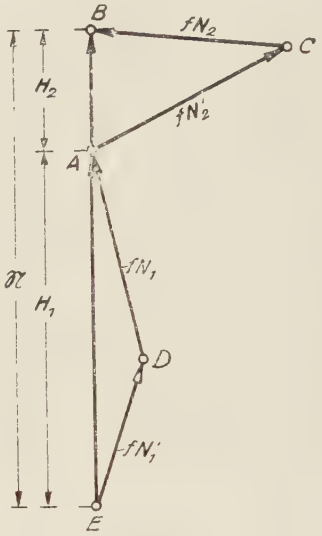


Fig. 23*a*.

The coned envelope generated by the rotation of the line TO always strikes the check rail. In the section, figure 23*b*, the line TO is inclined by $\varepsilon = 7^0\ 40'$ against the vertical drawn as tangent to the check rail; $\beta = 90^0 - \varepsilon$ is therefore $= 82^0\ 20'$. In the side view, figure 23*c* the horizontal plane at the top-edge level of the check rail intersects, with the type 6 on steel, the circle T at a distance $d = \sqrt{11.1189} = 108$ mm. from the wheel contact point, and the circle O at a distance $d + x = \sqrt{27.1173} = 178$ mm. from the same point; for the types having the check rail flush with the running rail, O″ is at a distance $T''O'' = z = \sqrt{16.1184} = 137$ mm. from T″. The forward displacement is again calculated

as explained on page 784 by means of formula

$b = (r + a) \cdot \alpha \cdot \tan\beta = \sim r \cdot \alpha \cdot \tan\beta$;

$\tan\beta$ is very large = 7.43. For type 6 on

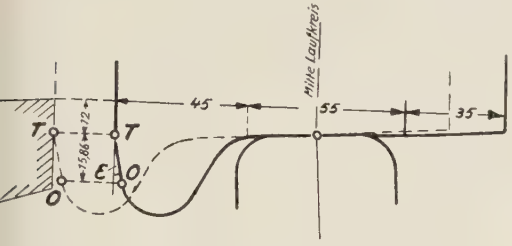


Fig. 23b.

Note. — Mitte Laufkreis = middle of tread circle.

steel (check rail 11-12 mm. above running rail), and wheels of 1.20 m. diameter, the point T′ strikes for α minimum up to $\alpha = 1°\ 23'$, with $b = 108$ mm. for $\alpha = 1°\ 23'$ up to $\alpha = 2°\ 16'$ the points of intersection T′O′.

For types having the check rail on a level with the running rail, the forward displacement reaches the point O″ already with $\alpha = 1°45'$. The forward displacement is always very large and amounts to 10-15 times the difference between the top surface of the running rail and that of the check rail. The frictional forces which, in figure 23*c*, arise at the points T′, O′, O″ during the rotation of the wheel around its momentary axis of rotation T″, occur therefore nearly vertically upwards in the plane of this figure and in O exactly vertically upwards; they have practically no component in the horizontal plane (rail plane), i. e. they hardly divert the force R_a from the check rail directing force $\mathfrak{N}$.

To prove this, we ascertain, according to figure 6, the direction of R_a for an unfavourably selected centre of friction M, in the case of the most unfavourable type

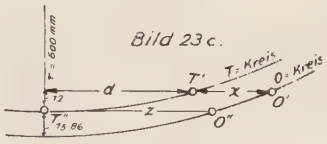


Fig. 23c.

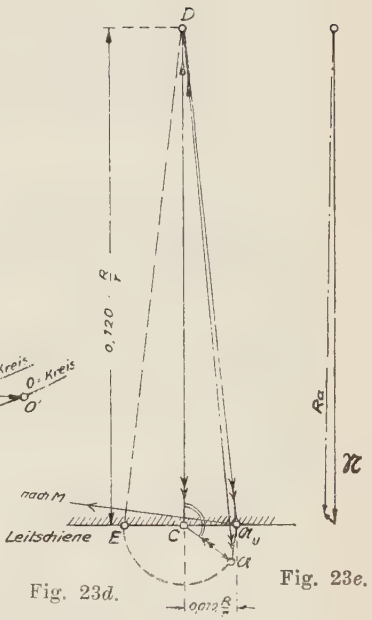


Fig. 23d. Fig. 23e.

Note. — Leitschiene = check rail. — Kreis = circle.

of check rail, type 6 on steel, the check rail top being above the running rail head (fig. 23*d*).

The point $\mathfrak{A}$ moves in the horizontal plane from $\mathfrak{A}_u$ to $\mathfrak{A}$, in the vertical plane from D to $\mathfrak{A}_u$, i. e. from D to $\mathfrak{A}$. If $\overline{\mathfrak{A}C}$ represents the horizontal projection of the total frictional force $\overline{\mathfrak{A}D}$ on the check rail, the normal pressure $\mathfrak{N} = \frac{\mathfrak{A}D}{f} = \frac{DE}{f}$. The direction of R_a is obtained by combining $f \cdot \overline{C\mathfrak{A}}$ with the length $\overline{DE}$, as shown in figure 23*e*. The direction of R_a deviates very little from the normal pressure $\mathfrak{N}$. The force $\mathfrak{N}$ acts against the wheel nearly perpendicular to the check-rail flange struck by the wheel; no error is committed if the force R_a is made to coincide in direction with $\mathfrak{N}$, if it is assumed to be parallel to the wheel direc-

tion, and if the frictional force at the striking point $\mathfrak{A}'$ is assumed directed vertically upwards. This vertical frictional force should then be taken as equal to $f \cdot \mathfrak{R} = \mathfrak{F}$.

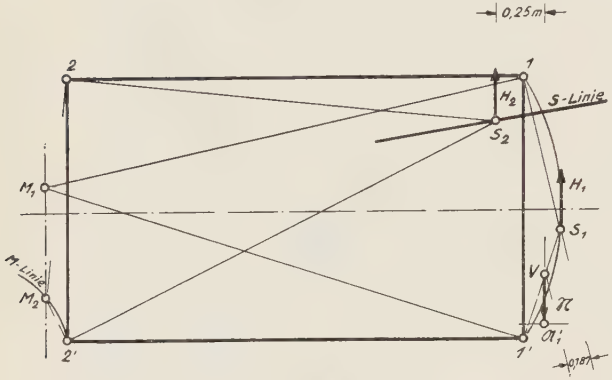


Fig. 23.

In the vehicle (fig. 23) with two carrying pairs of wheels the resultant of $fN_2 \overset{\wedge}{+} fN'_2 = H_2$ is perpendicular to the longitudinal axis of the vehicle, as the trailing wheels are supposed to run free; therefore, as $\mathfrak{R}$ is also perpendicular, the resultant of $fN_1 \overset{\wedge}{+} fN'_1 = H_1$ must also be perpendicular and, together with H_2, maintain the balance $\mathfrak{R} = R_a$. One glance at the location diagram shows that H_2 must be on our known S line (see fig. 4), close to the left of the force $\mathfrak{R}$, as H_1 is obviously close to the right of $\mathfrak{R}$. The distance of H_2 from axle II is approx. $l = 2.60$ m., to which corresponds an $H_2 = 0.14$ Q. M_{II} is on the M line (see fig. 4) and will be, even if H_2 has not been correctly selected, nearly exactly 12 cm. behind axle II. The correct value ascertained is $H_2 = 0.145$ Q.

The $H_2 = 0.145$ Q gives a moment relatively to point $\mathfrak{A}'_1$ of $0.145 \cdot 0.25 = 0.0363$ Qm. The centre of friction M_I must be so located that $fN_1 = \frac{1}{4} \cdot 1.058 = 0.264$ Q passes at a distance of $0.0363 : 0.264 = 0.137$ m. to the right of V_1, the point of intersection of $\mathfrak{R}$ and fN_1. If the frictional forces are combined in the force diagram according to their ascertained directions, the following values will be obtained from the diagram :

$fN'_1 = 0.198$ Q, $\mathfrak{R} = 0.592$ Q; $\mathfrak{F}$ (the frictional force at point $\mathfrak{A}'_1$, directed vertically upwards) becomes equal to $\frac{\mathfrak{R}}{4} = 0.148$ Q, and we get $N'_1 + f \cdot \mathfrak{R} = 0.792 + 0.148 = 0.940$ Q, which must be the case if the forces in the vertical plane are also to be in equilibrium i. e. $Q'_1 = 1.00 - H_2 \cdot \frac{0.6}{1.5} =$ in round figures 0.94 Q. The point of intersections S_1 of the frictional forces fN_1 and fN'_1 is located on the parabolic arc through the points 1 and 1′ (see fig. 2), which arc belongs to the M vertical and remains practically the same, whether the M vertical lies immediately on the axle II or a few centimetres behind it in the location diagram.

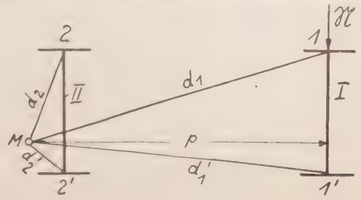
It should be noted here that the above analysis for a statically guided vehicle in check-rail track will only rarely find its application in the future, owing to the gradual introduction of the narrow gap (up to 50 mm.) between the check rail and the inside rail, as with a narrow gauge and this reduced distance nearly every vehicle will be geometrically guided.

E) Summary and discussion of the results for vehicles with rigid axles.

In the Table 2 hereafter have been summarised the values obtained from the force diagram and calculated for a series of investigations on different vehicles for all kinds of force conditions.

TABLE 2.

Number of axles	Wheelbase	Coupled.	PS	Z	S	H	N_1	N'_1	N_2	N'_2	$\mathfrak{N}$	$\mathfrak{N}_z + \mathfrak{T}$	$\frac{Q_1}{\mathfrak{N}_z + \mathfrak{T}}$	$N_1 d_1$	$N'_1 d'_1$	$N_2 d_2$	$N'_2 d'_2$	$\mathfrak{N} \cdot \eta \cdot d_1$ (*)	W (in Q)	w (kgr./t.)	p	$\frac{\Sigma R}{R}$	ΔR_I	ΔR_{II} / ΔR_{III}	Remarks.	Figure.
	metres																									
2	2.60	...	0	0	0	0.176	1.0	1.0	1.0	1.0	0.666	...	...	2.97	2.97	0.8	0.8	...	...	...	2.88	...	0	0	...	1*a*
2	2.60	no	0	0	0	0.145	0.575	0.942	0.942	1.058	0.675	0.485	2.19	1.79	2.58	1.15	0.35	3.93	0.01225	3.063	2.735	0.0049	0.54	0.45	...	10
2	2.60	no	▲ 0.2	0	0	0.064	0.56	0.863	0.997	1.003	0.818	0.613	1.86	1.70	2.34	0.76	0.76	4.64	0.01275	3.188	2.70	0.0058	0.65	0.0	...	
2	2.60	no	▼ — 0.2	0	0	0.225	0.61	1.018	0.887	1.113	0.560	0.372	2.63	1.90	2.95	1.065	0.498	3.26	0.01210	3.025	2.87	0.0041	0.46	0.41	...	11
2	2.60	yes	0	0	0	0.145	0.57	0.942	0.942	1.058	0.680	0.483	2.19	1.71	2.58	1.20	0.280	3.88	0.01206	3.015	2.73	0.00485	0.52			
2	2.60	yes	▲ 0.25	0	0	0.043	0.55	0.843	1.011	0.989	0.842	0.607	1.905	1.61	2.26	1.285	0.228	4.59	0.01247	3.118	2.63	0.00574	0.52		...	..
2	2.60	yes	0	0.1	0	0.146	0.56	0.925	0.961	1.078	0.687	0.480	2,12	1.645	2.80	0.784	0.807	4.06	0.01262	3.155	2.83	0.00507	0.03		...	14
2	2.60	yes	0	0.5	0	0.082	0.547	0.88	0.385	1.124	0.542	0.517	2.06	1.44	2.64	0	1.686	3.20	0.01121	2.803	2.60	0.00400	— 0.75			..
2	2.60	yes	0	0.85	0	+ 0.0094	0.61	0.83	1.16	1.16	0.386	0.23	3.65	1.79	3.18	0.615	2.32	2.55	0.01307	3.268	2.64	0.0032	— 2.00		...	15
2	2.60	yes	0	0.875	0	+ 0.014	0.59	0.83	1.17	1.17	0.357	0.25	3.36	1.85	3.42	1.95	3.70	2.46	0.01672	4.18	2.66	0.0031	— 2.42		...	..
2	2.60	yes	▲ 0.20	0.75	0	— 0.10	0.64	0.81	1.13	1.14	0.53	0.293	3.18	1.575	2.40	0.294	1.94	3.09	0.01163	2.91	2.43	0.00386	— 0.94		...	..
2	2.60	no	0	0.17 II	0	0.112	0.56	0.954	0.954	1.046	0.655	0.494	2.15	1.62	2.48	0	1.565	3.54	0.01151	2.88	2.60	0.0044	0.54	— 0.75	...	...
2	2.60	no	▼ — 0.187	0.17 II	0	0.205	0.55	0.988	1.137	0.929	0.537	0.496	1.90	1.60	2.60	0.023	1.35	2.90	0.0106	2.65	2.62	0.0036	0.51	— 0.73		
2	2.60	no	▲ 0.213	0.17 II	0	0.013	0.55	0.844	1.069	0.997	0.760	0.540	2.02	1.60	2.19	0	1.496	4.10	0.01174	2.935	2.60	0.0051	0.52	— 0.75		
2	2.60	no	0	0.35 II	0	0.059	0.607	0.977	0.977	1.023	0.590	0.416	2.51	1.75	2.55	0	1.535	3.22	0.01132	2.83	2.60	0.0040	0.48	— 0.75	without ΔQ_z.	
2	2.60	no	0	0.17 I	0	0.147	0.61	0.942	0.942	1.159	0.679	0.448	2.37	1.66	2.86	1.18	0.325	3.95	0.0125	3.125	2.72	0.0049	— 0.60	0.50	ditto	...
2	2.60	no	▼ — 0.260	0.17 I	0	0.283	0.60	0.987	0.880	1.186	0.530	0.347	2.71	1.78	3.32	0	0.646	3.45	0.0128	3.20	2.98	0.0043	— 0.87	0.37	...	..
2	2.60	no	▲ 0.187	0.17 I	0	0.047	0.535	0.868	1.057	1.009	0.732	0.531	2.00	1.39	2.55	1.04	1.514	4.22	0.0121	3.025	2.60	0.0053	— 0.64	0.75	ditto	...
2	2.60	no	0	0.35 I	0	0.135	0.66	0.946	0.946	1.054	0.43	0.394	2.67	2.03	3.77	1.23	0.238	2.88	0.0126	3.15	2.70	0.0036	— 2.22	0.56	without ΔQ.	...
2	2.60	yes	▲ 0.25	0	0.125	0.01	0.45	1.00	1.00	0.560	0.81	0.55	1.81	1.35	2.60	1.50	0	4.30	0.0126	3.015	2.60	0.0054	0.75		ditto	16
2	2.60	yes	▲ 0.20	0	0.34	— 0.038	0.62	1.004	0.946	0.315	0.75	0.568	2.10	1.86	2.61	1.42	0	3.95	0.0123	3.075	2.60	0.0049	0.75		without ΔQ_z.	
2	2.60	yes	▲ 0.175	0	0.75	— 0.066	0.81	1.068	0.912	0.800	0.613	0.41	3.0	3.06	2.92	2.75	1.23	2.70	0.0158	3.95	2.27	0.0034	2.25			...
2	2.60	no	0	0	0	0.145	1.058	0.79	0.942	1.058	0.59	0.148	6.36	2.95	2.27	1.19	0.296	23.4	0.0376	9.40	2.72	0.0293	— 0.11	0.51		23
2	2.60	no	▼ — 0.260	0.170 I	0	0.272	0.942	0.88	0.885	1.181	0.425	0.106	9.35	2.84	3.19	1.045	0.614	16.8	0.0306	7.65	2.96	0.0210	— 1.34	0.38		
2	2.60	no	▼ — 0.187	0.170 II	0	0.208	0.946	0.86	0.921	1.137	0.455	0.114	8.65	2.59	2.38	0.046	1.672	18.0	0.0225	5.63	2.65	0.0225	— 0.06	— 0.72	Check-rail track.	
2	2.60	no	▲ 0.213	0.170 II	0	0.010	1.100	0.67	0.335	1.013	0.655	0.164	5.1	2.91	1.84	0	1.52	26.0	0.0403	10.07	2.60	0.0325	— 0.18	— 0.75		
2	2.60	no	0	0.35 II	0	0.060	0.957	0.79	0.45	1.091	0.48	0.12	7.6	2.58	2.15	0	1.635	18.7	0.0313	7.83	2.60	0.0234	— 0.11	— 0.75		
2	6.20	no	0	0	0	0.051	0.59	0.98	0.98	0.755	0.607 (I) 0.435 (II)	0.41	2.49	3.31	5.34	1.25	0.665	6.80 (I) 1.74 (II)	0.0239	5.98	5.45	0.00835 (I) 0.0022 (II)	0.574	0.288	R = 500 m.	20
3	4.50	no	0	0	0	0.095	0.60	0.961	0.645 (II) 0.960 (III)	1.00 (II) 1.039 (III)	0.616 (I) 0.646 (II)	0.424	2.45	2.81	4.34	1.643 (II) 1.44 (III)	2.28 (II) 0.0 (III)	5.02 (I) 3.05 (II)	0.0257	4.29	4.50	0.0063 (I) 0.0038 (II)	0.54	0.44 (II) 0.75 (III)	Middle axle strikes.	17/18
3	4.50	no	0	0	0	0.465 (II) 0.131 (III)	0.525	0.866	0.809 (II) 1.052 (III)	1.191 (II) 0.948 (III)	0.900	0.610	1.86	2.42	3.82	1.893 (II) 0.254 (III)	2.66 (II) 1.244 (III)	7.92	0.0253	4.21	4.40	0.0099	0.59	0.17 (II) — 0.56 (III)	Middle axle does not strike.	19
2	2.60	yes	0	0	0	0.118	0.635	0.953	0.953	0.950	0.637	0.423	2.47	1.905	2.48	1.43	0	4.72	0.00527	1.32	2.60	0.00236	0.75		R = 500 m.	..
2	4.50	yes	0	0	0	0.080	0.595	0.568	0.968	1.000	0.625	0.442	2.34	1.785	2.520	1.45	0	3.44	0.0114	2.85	2.60	0.0043	0.61		R = 500 m.	...



(*) $\eta = \frac{\overline{DE}}{\mathfrak{N}\ \mathfrak{N}_u}$ (see pp. 788/9).

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Result :

Table 2 and the drawings show that :

(1) As regards the force $\mathfrak{N}$ (normal wheel flange pressure) :

An outwards acting transverse force increases $\mathfrak{N}$ considerably.

An inwards acting transverse force markedly decreases $\mathfrak{N}$.

Tractive and braking forces decrease $\mathfrak{N}$.

Check rails decrease $\mathfrak{N}$.

(2) For short vehicles striking only at the front, the centre of friction moves up and down close to the rear-axle, according to the presence and influence of outside forces; tractive efforts push it very quickly, inwards acting transverse forces push it very slowly towards the bearing point of the rear outside wheel; medium tractive efforts push it towards the same point and fix it there, and only very large tractive efforts push it sideways and outwards beyond this bearing point; braking forces displace the centre of friction very rapidly, outwards acting side forces fairly rapidly towards the bearing point of the rear inside wheel; large braking forces displace it still further inwards beyond the inner rail. Therefore, only for very large tractive and braking forces, shortly before the wheels slip or pick up, is the centre of friction forced outside the rails.

In the case of fairly long vehicles (striking only at the front), and on curves of fairly large radii, the centre of friction is transferred to the inner rear wheel without intervention of any additional outside forces.

(3) The frictional work of the wheel flange at point $\mathfrak{A}$, i. e. at the inside of the outside rail ($\mathfrak{A}_R$) increases markedly with outwards transverse forces; it decreases markedly with inwards transverse forces; it also decreases with tractive and braking forces, but to a smaller extent. $\mathfrak{A}_R$ increases proportionately to the distance of the centre of friction M from the striking wheel. As generally with narrow gauge the line of the friction centres (main radius) advances nearer to the front, i. e. towards axle I, $\mathfrak{A}_R$ is considerably reduced (even for larger wheel bases), and consequently the wear of the struck outside rail is also far more reduced than for wide gauge.

The frictional work in check-rail track is very large indeed, and the curve resistance is a multiple of that for a track without check rails.

F) Determination of the forces exerted on vehicles with free radial axles.

In his paper : « Das Geheimnis der freien Lenkachsen » (The Secret of the free radial axles) *(Z. V. d. E.* No. 14, 1929), Dr. Bäseler has analysed the behaviour of the free radial axles by means of an example with an assumed two-point contact.

It is possible for every free radial axle to be oblique relatively to the perpendicular position to the longitudinal axis of the vehicle (tensionless position); because of the tendency of the bearing springs to return the axle into its perpendicular position, it exerts a force moment on the frame, which, in the above-mentioned paper, is given, for a definite spring arrangement, as :

$$\mathfrak{M}_z = 0.0734 \left(\frac{1}{\text{cm}}\right) \cdot Q \cdot \Delta\, l\ (\text{cm.}) \cdot s'\ (\text{m.})$$

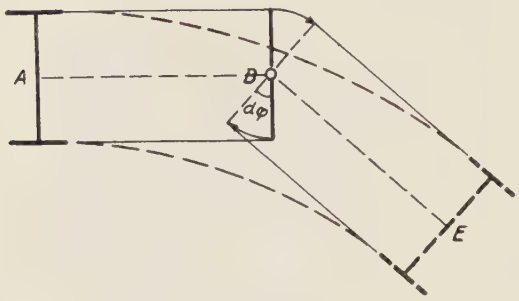


Fig. 24a.

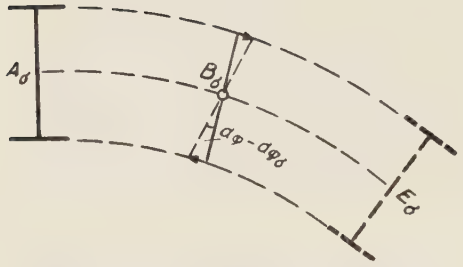


Fig. 24b.

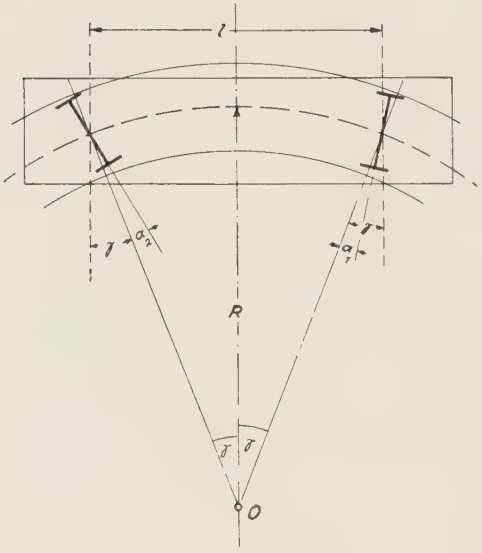


Fig. 25.

wherein Δl (in cm.) stands for the small displacement of the axleboxes relatively to the carriage body, and s' (in metres) is the distance between the two spring centres.

If γ is the angle formed by the longitudinal axis of the vehicle with the curve in the case of both pairs of wheels striking outside (see fig. 25 corresponding to fig. 19 of BÄSELER's paper), and α the striking angle of the axle placed obliquely, then we get : $\Delta l = (\gamma \pm \alpha) \cdot \frac{s'}{2}$ (assuming $s' = 194$ cm)

The centralizing moment is, therefore

$$\mathfrak{M}_z = 0.0734\,Q \cdot (\gamma \pm \alpha)\,\frac{194}{2} \cdot 1.94\,\text{m.t.}$$

$$\mathfrak{M}_z = (\gamma \pm \alpha) \cdot 13.8 \text{ in } Q \cdot m \text{ units.}$$

For a given position of the vehicle, e. g. for the geometrically guided vehicle, γ is known, and α to be found. This is very easy when the conic shape of the tyres is taken into account. With conical wheel tyres, the pure rolling motion (not taking part in the frictional work) no longer takes place along a straight path, but on a curve which, for an outwards striking axle, is curved in the same direction as the track curve.

When a pair of wheels, fitted with conical tyres, and with a side play σ of the wheels in the track and, therefore, a displacement of $\frac{\sigma}{2}$ relatively to the centre line of the track, rolls on a curve of radius R_σ, these wheels, when running along a track curve of radius R, behave exactly in the same manner as if it they moved through a curve of radius $R_e = \frac{R \cdot R_\sigma}{R_\sigma - R}$ (as long as $R_\sigma > R$). A portion of the lateral displacement is already converted into pure rolling by the conicity.

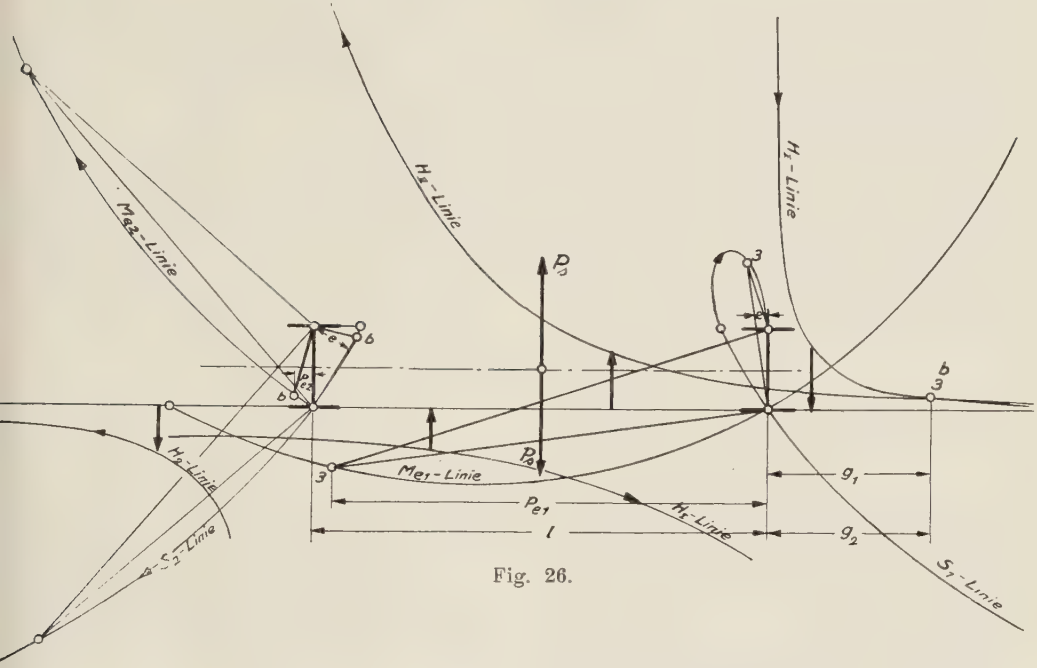


Fig. 26.

The formula of R_e is obtained as follows: When a pair of wheels with cylindrical tyres moves along a curve of radius R (fig. 24*a*) from A towards E, the motion can be considered as if the wheels would roll straight on as far as B, then describe, while slipping, an angle $d\varphi = \frac{AE}{R}$, and subsequently roll in this new position as far as E. The outwards striking pair of wheels with conical tyres, however, would move in pure rolling motion on the curve R_σ from A_σ to B_σ (fig. 24*b*) and should be compelled to rotate there, while slipping by the angle $d\varphi - d\varphi_\sigma$, and then roll on the new curve R_σ as far as E_σ. We have here $d\varphi_\sigma = \frac{1}{R_\sigma}$ and $d\varphi - d\varphi_\sigma = \frac{1}{R} - \frac{1}{R_\sigma} = \frac{R_\sigma - R}{R \cdot R_\sigma} = \frac{1}{R_e}$.

The conical shape of the wheel tyres has been taken into account in the analysis below.

With $r = 50$ cm., and $\sigma = 31$ mm., we get $R_\sigma = \frac{20 \cdot r \cdot s}{\sigma} = 484$ m. and $R_e = 790$ m.; for $\sigma = 41$ mm., we get $R_\sigma = 366$ and $R_e = 1645$ m.

We shall proceed to the graphical calculation for a vehicle with two free radial axles, having $l = 8.40$ m., and $\sigma = 31$ mm., R = 300 m., i. e. such as assumed by Bäseler in his paper mentioned above. As vehicles with free radial axles have always a large wheelbase, H becomes small and its influence on the wheel pressures is so insignificant, that it can be overlooked, i. e. the wheel pressures of the non-striking wheels can be assumed = Q. As Bäseler has proved, in his example both axles strike the outer rail, which we have also found to be exact. However, the question as to whether, when the vehicle is in the state

of inertia, the rear axle will strike the outer or inner rail depends on the centralizing force of the carriage spring suspension, the radius of the curve, the wheelbase, and also on the gauge of the track and other factors. If the restoring force is low, the rear axle is certain to strike the outer rail. If the angle γ is large, i. e. if the ratio of the wheel base l to the curve radius R is large, the rear axle will no longer strike the outer rail. With the vehicle considered here (with $l = 8.40$ m. and the spring arrangement and centralizing effect mentioned above), the rear axle hardly strikes the outer rail for $R = 230$ m., and not at all if the curve radius ($\sin \gamma < 0.0183$) is smaller. The construction shall be carried out as if the vehicle were to run through a curve of $R_e = 790$ m. The distances p of the centres of friction from the axles will also be given the index e, and will consequently be p_{1e} and p_{2e}. The following formula applies for axle I :

$$\mathfrak{M}_{z1} = (\gamma - \alpha_1) \cdot 13.8\ Qm \text{ and } p_{1e} = \alpha_1 \cdot 790.$$

For axle II the formula becomes :

$$\mathfrak{M}_{z2} = (\gamma + \alpha_2) \cdot 13.8\ Qm \text{ and } p_{2e} = \alpha_2 \cdot 790.$$

In table 3 below, the corresponding $\mathfrak{M}_z$ and p_e are tabulated for some α_1 and α_2. Knowing the distance e (see sketch under table 3) to be $\mathfrak{M}_{z1} : fN'_1$ and $\mathfrak{M}_{z2} : fN'_2$ from the striking wheel ($e = \mathfrak{M}_z : fQ$) the direction of the frictional forces fN'_1 and fN'_2 are obtained at the same time; similarly, knowing the point of intersection with the perpendiculars to these forces at the distance p_e, we get immediately the corresponding centres of friction M_e. The H corresponding to each α are found again exactly according to figure 9*a*. Some of them are given in Table 3. As regards the distance g_1 or g_2 from axle I, at which act the forces replacing $\mathfrak{M}_{z1}$ and H_I, and $\mathfrak{M}_{z2}$ and H_{II} respectively, i. e. $(-H_I)$ and $(-H_{II})$, it can be found by means of the force diagram, but it can also be calculated by means of the formula $\mathfrak{M}_{z1} = g_1 \cdot H_I$, or $\mathfrak{M}_{z2} = (l + g_2) \cdot H_{II}$.

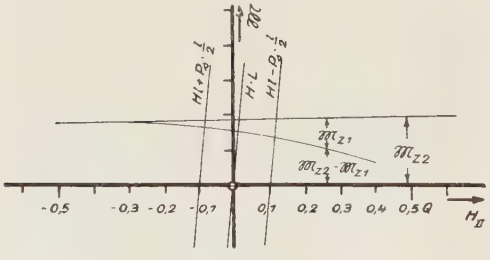


Fig. 27.

Figure 26 shows how, with increasing α, the centres of friction M_e, the point of intersection S of the forces fN, and the values H and g vary.

In figure 27, $\mathfrak{M}_z$ is plotted in terms of H. The point of intersection of the $H \cdot l$ line with the $\mathfrak{M}_{z2} - \mathfrak{M}_{z1}$ line gives the correct H, $\mathfrak{M}_{z1}$ and $\mathfrak{M}_{z2}$, and thereby, all other values. The centres of friction of the axles are the points 3 and b on the M lines. Figure 28 shows the force diagram for the complete vehicle.

From it can be read : $H_I = -H_{II} = 0.0174$ Q. $\mathfrak{M}_{z1}$ becomes 0.0516 and $\mathfrak{M}_{z2} = 0.199$.

If now an inwards directed side force $P_s = 0.2$ Q acts in the centre of the vehicle, we have :

$$\mathfrak{M}_{z2} - \mathfrak{M}_{z1} = H_{II} \cdot l + P_s \cdot \frac{l}{2}$$

In figure 27, it is enough to displace the H . l line by a quantity $\frac{P^s\, l \cdot}{2 \cdot l} =$ 0.1 Q, in order to obtain at once :

$$H_{II} = -0.1143\ Q,\ \mathfrak{M}_{z2} = -0.210,\ \mathfrak{M}_{z1} = 0.090\ Q.\ m$$

If $P_s = 0.2$ Q acts outwards the H . l line must be displaced to the other side by 0.1 Q, giving :

$$H_{II} = 0.082\ Q,\ \mathfrak{M}_{z2} = -0.190$$
$$\text{and } \mathfrak{M}_{z1} = 0.035\ Q.\ m.$$

We have drawn in the location diagram (fig. 26), the two resultants $H_{II} = -0.1143$ and $+0.083$, $H_I = -0.0857$ and $+0.118$ obtained for the two examples of $P_s = 0.2$ Q acting inwards, and $P_s = 0.2$ Q outwards. It will be seen that these resultants balance the P_s.

The curve resistance is again calculated as in Chapter C 3, according to the formula :

$$w = \frac{\Sigma f N \cdot d + \Sigma \mathfrak{N} \cdot f \cdot r_1 \cdot d}{R_e} \cdot \frac{1\,000}{n}$$

which gives $w = 2.35$ kgr./t. for the vehicle in question, if it is not submitted to transverse forces. This result agrees, of course, exactly with the values calculated by Bäseler.

If, however, a larger σ is assumed, i. e. a larger play between wheel and rail, say $\sigma = 41$ mm., w then will become very small because the conicity of the tyres makes such a vehicle run nearly frictionless round the curve, e. g., for a vehicle of 6.00 m. wheelbase, the resistance is only $w = 0.97$ kgr./t.

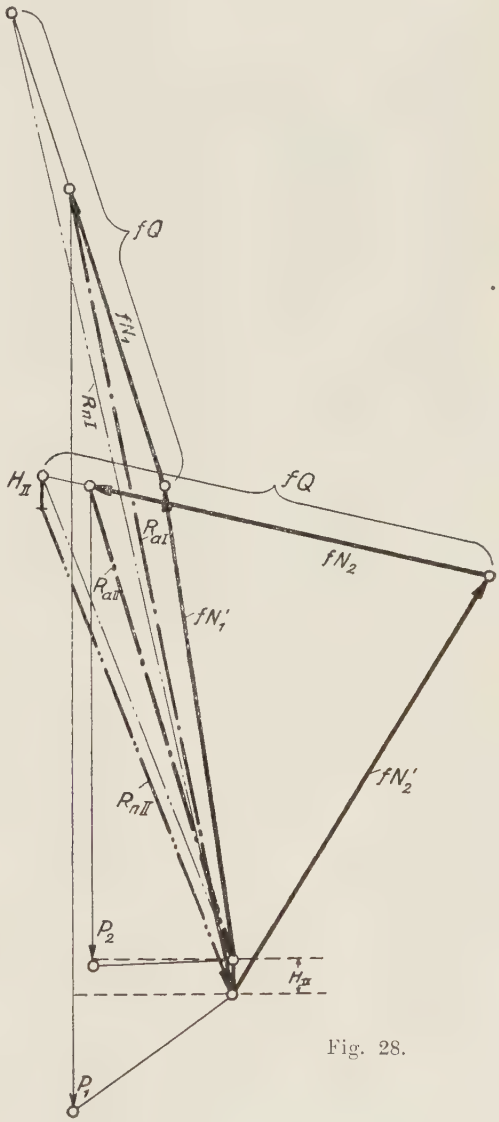
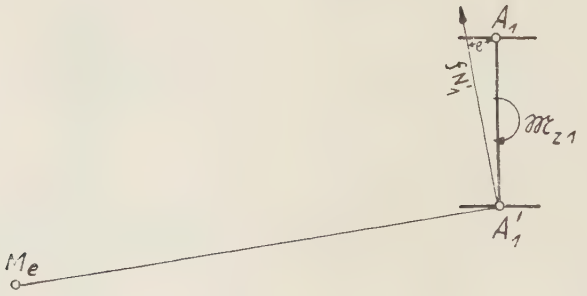


Fig. 28.

TABLE 3.

1	2	3	4	5	6	7	8
2	α_1	$\gamma - \alpha_1$	$\mathfrak{M}_{z_1}$	e_1	p_{e_1}	p_1	H_I
3	...	...	...	...	...	...	...
4	0.0000	0.014	0.193	0.773	0.00	0.00	0.95
5	0.00333	0.01067	0.147	0.589	2.63	1.00	0.34
6	0.00700	0.00700	0.0966	0.386	5.53	2.1	...
7	0.01027	0.00373	0.0516	0.2064	8.11	3.08	0.0174
8	0.01333	0.00067	0.00928	0.0372	10.55	4.0	...
9	0.01400	0.0140	0.00	0.0	11.05	4.2	— 0.30
10	α_2	$\gamma + \alpha_2$	$-\mathfrak{M}_{z_2}$	e_2	p_{e_2}	p_2	H_{II}
11	0.000133	0.014133	0.195	0.78	0.1315	0.050	+ 0.05
12	0.00039	0.01439	0.199	0.795	0.305	0.116	— 0.0174
13	0.00167	0.01567	0.216	0.866	1.315	0.50	— 0.81
14	0.00333	0.01733	0.239	0.97	2.63	1.00	...
15	0.00667	0.02067	0.285	1.14	5.26	2.00	...



[625. 254]

Air conditioning in railway carriages,

by Mr. PLA,
Ingénieur à l'Office central d'Études de Matériel de chemin de fer (France).
(*Électricité*, March and May, 1937.)

Air conditioning in railway carriages raises different problems than when applied to buildings.

The cubic space available per occupant is so extremely low — hardly exceeding, in fact, 1 m³ (35 cu. ft.) in 3rd-class compartments and not reaching 2 m³ (70 cu. ft.) in 1st-class accommodation — that the renewal of air, whilst being relatively rapid, must not be such as to cause uncomfortable draughts, in spite of the fact that the air inlets and outlets must of necessity, owing to the restricted size of compartments, be situated in close proximity to the passengers. Further, the influence of the passengers themselves is a factor of much greater relative importance than in the case of large halls.

The thermal transmission to and from the exterior through the walls of the compartment is also appreciable, however much attention may be paid to their insulation. With metal stock, in particular, the insulation factor is of great importance.

The fact that the exterior air is laden with dust in the neighbourhood of the ballast and polluted with smoke higher up adds to the difficulties of filtration, already aggravated by the lack of sufficient space for the installation of large washing chambers.

Lack of space also creates limiting conditions for the installation of the air-conditioning plant as a whole. It must be as *light* as possible, for with the railway as with all means of transport, weight is costly.

Another important factor is the relatively large amount of *energy* required for working the installation. On electrified lines a practically unlimited amount of energy is available in a convenient form and at low cost, but such is not the case with steam-hauled trains nor with self-propelled units. On steam trains, energy is available either in the shape of steam or in mechanical form through a drive from the axles, or else from an independent motor. We shall examine how all these forms of energy are utilised in the various air-conditioning systems employed.

In a railway carriage, as in a fixed installation, a complete system of air conditioning includes :

(1) Fans and ducts for the circulation and distribution of air;

(2) The air-conditioning unit itself, in which the air is filtered, warmed or cooled, humidified or dried;

(3) A source of heat;

(4) A source of cold;

(5) A source of energy;

(6) Indicating and regulating apparatus.

Some climates do not necessitate so complete an installation. In certain hot countries, for instance, there is no need for any source of heat, whilst in the temperate climates the cold-producing element can be dispensed with, or recourse can be had to simple apparatus for the evaporation of water, either naturally or artificially accelerated by forced currents of air. On the other hand, in hot and humid countries a refrigerating plant of some capacity is essential, and extreme climates call for a complete installation, with in most cases additional heating equipment in-

dependent of the air-conditioning plant, as the air flow that would be required to supply the amount of heat desired would be far too rapid.

We will examine successively how the problem has been, or is being, solved in the various countries, and in doing so we shall meet with most of the possible combinations.

The first attempt at air conditioning in France was made in 1929 by the *Paris-Orléans Railway* which equipped a carriage with a plant for supplying air cooled by passage over ice. The ice bunker was composed of three compartments between which the air circulated. The ice water remaining in the bunker ensured transmission between the ice and the metal walls. This carriage was shewn at the Colonial Exhibition of 1931.

American air-conditioning systems.

It was in the United States, however, that air conditioning in trains was originally developed, and it is the only country where the practice is widely extended. The first air-conditioned carriages were equipped in 1929, and the existing stock is now being converted at the rate of several thousands a year (3 349 in 1935).

Method of air-distribution.

American passenger stock is not, generally speaking, of the compartment type, and the air is distributed near the ceiling by one or other of the following two methods :

— In the « bulkhead system » (fig. 1) there is one, or more often two, air-conditioning units located over a false ceiling at one or both ends of the carriage. After suitable treatment the air is blown parallel to the ceiling, forming a layer extending over its whole length and only partially mixing with the surrounding air, which remains completely undisturbed in the zone occupied by the passengers.

— In the « duct system » the air proceeding from the conditioning plant is propelled along two longitudinal ducts situated on either side of the ceiling and opening into the various outlets distributed in the coach.

The first system is less costly, easier to install, and the power required for the fans is only 1/7 to 1/4 H. P. as compared with 3/4 to 1 H.P. for the second.

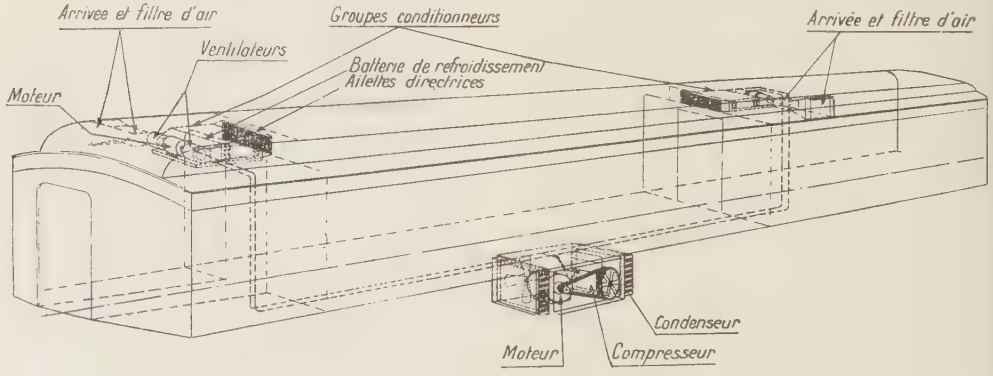


Fig. 1.

Explanation of French terms:

Ailettes directrices = deflector vanes. — Arrivée et filtre d'air = air intake and filter. — Batterie de refroidissement = cooling plant. — Compresseur = compressor. — Condenseur = condenser. — Groupes conditionneurs = air-conditioning units. — Moteur = motor. — Ventilateurs = fans.

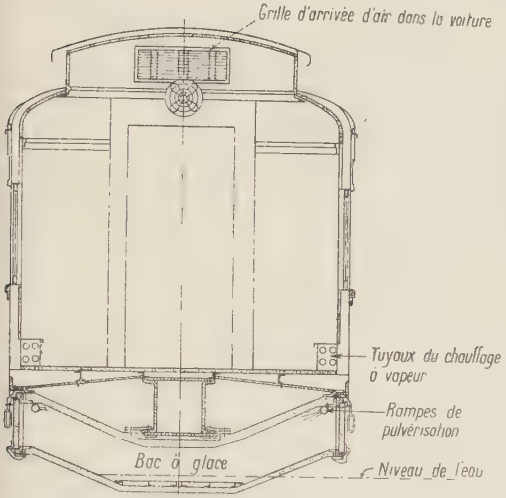


Fig. 2.

Explanation of French terms:
Bac à glace = ice bunker. — Grille d'arrivée = inlet grill for air supply to carriage. — Rampes de pulvérisation = spray nozzles. — Niveau de l'eau = water level. — Tuyaux de chauffage à vapeur = steam-heating pipes.

On the other hand, air distribution control is less easy, and the system can not be employed on compartment stock. Further, as the heating capacity of the plant is not sufficient in very cold weather, the normal steam-pipe heating equipment with pipes running along the sides of the coach near the floor has been retained. Both systems are regulated, depending on the installations, either manually by the train attendants or automatically by a thermostat.

Air-conditioning units.

As previously stated, the conditioning units are installed over a false ceiling at one or both ends, and consist of the following :

— An air intake from the exterior, with viscous or wool filter;

— An air extractor inside the coach. The re-circulated air is not filtered except in the case of smoking and restaurant carriages. The proportion of fresh air supplied varies in general from 25 to 50 % ;

— The air circulation fans are restricted to two for reasons of space, and are mounted at each end of a single motor which is designed for 32-v. d. c. and often for three speeds, the rating varying from 1/6 to 1/4 H.P.;

— The cooling plant consists in most cases of finned copper tubes, each unit having a cooling surface of 20 to 30 m² (215 to 323 sq. ft.), or 40 to 60 m² (430 to 646 sq. ft.) per coach. The total cooling capacity varies from 8 000 to 15 000 calories (31 000 to 59 500 B. T. U.) per hour;

— The heating plant is similarly formed, with a heating surface in the neighbourhood of 7 m² (75 sq. ft.) and a capacity of 20 000 calories (79 000 B. T. U.) per hour.

In the « bulkhead » system the conditioned air is supplied through a grill with deflector vanes for directing the air distribution. In the « duct » system the fans are sometimes placed after the heating and cooling surfaces and blow direct into the distribution ducts (see fig. 4).

Humidification is rarely practised in America, and is effected by means of steam. De-humidification is obtained by condensation on the cold surface, and the conditioning unit includes a drain pipe for the condensate.

Sources of heat.

The source of heat is always steam from the locomotive, drawn from the steam pipe running throughout the train.

Sources of cold.

It is the source of cold that distinguishes the three main systems, which fall into three groups, viz.

Refrigeration by means of ice;

Refrigeration by water evaporation in

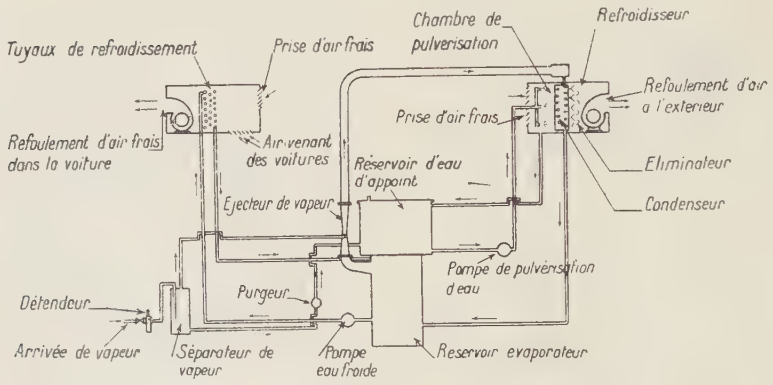


Fig. 3.

Explanation of French terms:

Air venant des voitures = air from compartments. — Arrivée de vapeur = steam supply. — Chambre de pulvérisation = water-spray chamber. — Condenseur = condenser. — Détendeur = pressure-reducing valve. — Ejecteur de vapeur = steam ejector. — Eliminateur = eliminator. — Pompe eau froide = cold water pump. — Pompe de pulvérisation d'eau = water-spray pump. — Prise d'air frais = fresh air intake. — Purgeur = drain cock. — Refoulement d'air frais = fresh air supply to compartment. — Refoulement d'air à l'extérieur = air outlet to atmosphere. — Refroidisseur = refrigerator. — Réservoir évaporateur = evaporating chamber. — Réservoir d'eau d'appoint = make-up water tank. — Séparateur de vapeur = steam separator. — Tuyaux de refroidissement = cooling pipes.

a vacuum produced by a steam ejector; Refrigeration by compressor plant.

Refrigeration by means of ice. — This system (fig. 2) consists of an ice bunker installed under the floor of the coach and carefully insulated. The ice is replenished from either side through double-walled doors. The capacity of these bunkers varies from 1 200 to 2 500 kgr. (2 600 to 5 500 lb.). The ice is sprayed with water, which collects in an ice-cooled condition at the bottom of the bunker from which it is extracted by one or more 1/4 H.P. centrifugal pumps and passed through a filter to the conditioning unit or units. Any excess of water in the bunker caused by melting of the ice is drained either direct to the outside of the coach or via an auxiliary cooling unit.

In this system the use of motive power is reduced to a minimum, the energy necessary for driving the fan and pump motors being supplied by the lighting generator of 4 to 5 kw., and a 500-a.h. battery of accumulators.

Refrigeration by steam ejector. — In this system the water is used as the cooling agent, the plant (fig. 3) being composed of an evaporator, an ejector and a condenser. A partial vacuum is produced in the evaporator by the ejector, by means of steam from the train pipe. The steam pressure is reduced to a constant value of 3.5 kgr./cm² (50 lb. per sq. in.) and passes through a water separator and regulating valve before entering the ejector, after which it condenses together with the steam from the evaporator in a tube condenser cooled by a water spray and forced draught from a special fan.

Circulation of cold water between the evaporator and the conditioning unit is maintained by a pump.

It is estimated that with the plant working at normal capacity, represented by the melting of 5 tons of ice per 24 hours, the consumption of steam at 3.5 kgr./cm² (50 lb. per sq. in.) is 80 kgr. (176 lb.) per hour. The motors for driving the fans and pumps absorb a total energy of 2.4 kw.

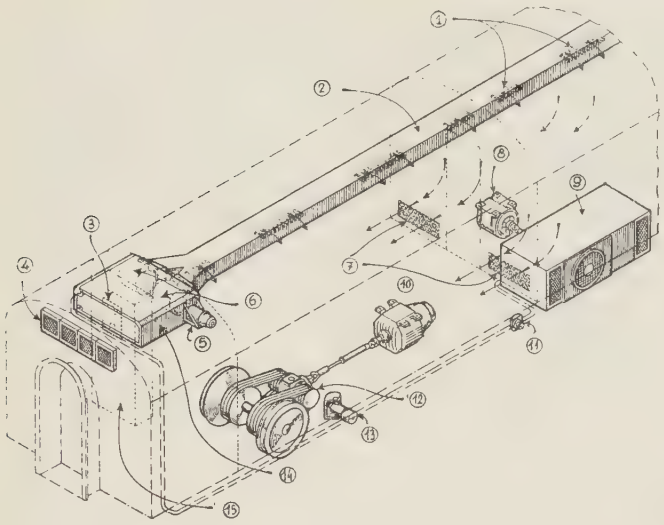


Fig. 4.

Legend:

1. Air distribution grills.
2. Conditioned-air supply duct.
3. Control panel.
4. Viscous oil filters.
5. Fan motor.
6. Fans.
7. Air intakes for closed-circuit circulation.
8. Pre-cooling motor.
9. Refrigerator unit.
10. Generator.
11. A. c. pre-cooling plug and socket.
12. Axle drive to generator.
13. D. c. pre-cooling plug and socket.
14. Air-conditioning unit with evaporator.
15. Cupboard for contactors.

Refrigeration by compressor plant. — This method of refrigeration is the most widely used. As with the other systems, the conditioning units are installed in the body of the coach and the refrigerating set below the underframe. The refrigerating set includes a compressor and a condensing plant consisting of finned tubes cooled by a special fan. The evaporator forms the cooling section of the conditioning set.

In the early installations the cooling agent used was ammonia and then methyl chloride. Today freon (dichlorodifluoromethane) is used almost universally. This substance boils at —8° C. (17° F.) and is odourless, and although practically non-toxic the Americans have perfected leak detectors based on the coloration of a flame in the presence of freon. As a gas it is stable, non-explosive and does not attack the metals used in the various parts of the plant. The air-cooled compressor is generally of the four-in-line cylinder type. The one used on the *Pennsylvania,* for instance, has four 80 × 80 mm. (3 5/32 in. × 35/32 in.) cylinders and runs at 425 r.p.m.

The types of drive used are somewhat varied. In the early days an independent petrol engine was employed. This practice was discontinued, but an independent motor running on propane is still met with occasionally. The propane is stored in liquid form in cylinders, and a total weight of 280 kgr. (620 lb.) of fuel plus cylinders will keep an 18-h.p. compressor running at full speed for 60 hours.

The power unit usually employed is an electric motor with vee-belt drive. This motor also operates the condenser

fan, the complete unit being mounted in an independent frame and installed below the vehicle underframe with special devices to prevent the transmission of vibrations. The motor is a 10-H.P. unit running on direct current at 36 or 72 volts from a generator driven off the axle or from a battery of accumulators of 1 000-a.h. capacity. To enable the vehicle to be pre-cooled or the plant to operate during long stops, a dual motor is used, the a.c. component of which is designed for current at 220 v., 60 cycles, and a rating of 15 H.P. (fig. 5).

The capacity of the generator for supplying power to the motor, fans and pumps, which is independent of the 4-k.w. lighting dynamo, is 15 or 20 k.w., and the transmission of this power from the axle raises difficult problems. In some cases the dynamo is fixed to the vehicle underframe and is driven by a cardan shaft with splined coupling in order to allow for movement of the bogie relatively to the vehicle body. The actual drive from the axle is by vee-belts to a gear box mounted on one of the bogie bolsters, and thence via the cardan shaft to the generator (fig. 4).

In some installations the generator is mounted on the bogie like a « nose-suspended » traction motor, and is connected direct to the axle through a train of gears. A safety friction clutch is sometimes incorporated in order to protect the dynamo from any damage that might be caused by sudden braking or starting of the vehicle.

The weight of the complete refrigerator plants, including the compressor-condenser unit, the evaporator unit, the regulation gear and the generator, varies from 2 to 2.5 tons for a capacity of 7 tons of ice per 24 hours.

In still another system, employed in particular by the *Pullman Car & Manufacturing Corporation*, the generator and the compressor motor are eliminated, the compressor being driven by mechanical transmission direct from the axle, with electrical speed control. The latter consists of an armature coupled by a train of gears to the axle, and a revolving field attached to the compressor. Variation of the field current causes a corresponding degree of slip between the armature and the field, thus maintaining the compressor at a practically constant speed, whatever the speed of the vehicle and the armature. When the vehicle is stationary the compressor is driven by a special 220-v. a.c. motor.

All these different systems of refrigeration have their advantages and disadvantages, and none seems to have been discarded altogether.

The ice-cooling system is obviously the simplest and the least costly to install, but with a full charge of ice it is relatively heavy and on long runs the ice has to be replenished one or more times en route, which means a serious addition to the servicing. Its advantage is that during periods when cooling is not required, the transport of heavy and bulky plant is avoided, as there is no need for any very powerful source of motive power. The system is thus eminently suitable for climates where very hot days are rare and where there are no very long runs.

The ejector system is simple and reliable in operation, does not necessitate any considerable amount of motive power other than steam, has few moving parts and requires no special supplies (neither ice nor cooling fluid is needed). On the other hand it is not suitable for use on long trains as it requires steam at a pressure of 3.5 to 4 kgr./cm^2 (50 to 57 lb. per sq. in.), which is difficult to obtain at the back of long trains.

The compression system is the most adaptable and ideal method, but also the most complicated and costly, chiefly owing to the large increase in power plant required.

As regards the cost of installation and subsequent maintenance, no figures can

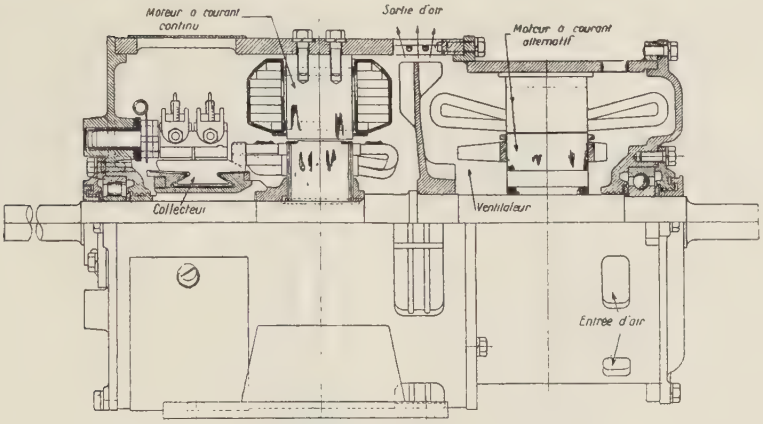


Fig. 5.

Explanation of French terms:

Collecteur = commutator. — Entrée d'air = air intake. — Moteur à courant alternatif = a. c. motor. — Moteur à courant continu = d. c. motor. — Sortie d'air = air outlet. — Ventilateur = fan.

be given in respect of the ice-cooling system as the cost of ice is extremely variable. In the case of the other two systems the figures quoted are 0.023 dollar per mile for the ejector system and 0.04 dollar per mile for the compression system.

The relative popuplarity of the various types of equipment to which reference has been made is shewn by the following table giving the respective numbers of vehicles equipped in the United States during 1935.

Refrigeration by means of :

Ice	989
Steam ejector	549
Electrically operated compressor	705
Mechanically operated compressor	1 160
Total	3 349

In the compressor systems the cooling agent employed is freon, except in the case of 12 vehicles on which methyl chloride is used.

Control and regulation.

The controlling and regulating arrangements are very varied, and attain great complexity in the case of the compressor installations incorporating a dual electric motor.

Temperature control is obtained by means of thermostats of the mercury-column type installed in the coach. There are generally three of these, set at different temperatures, e.g., 23°, 25.5° and 28° C. (73, 78 and 82° F.), one or other being switched into the circuit by the train attendant according to the season and time of day, and operating by making or breaking a control current which, by means of relays, starts up or stops the compressor motor. The latter cannot run unless the fans are working. The fan motors are fitted with single or three-speed control.

On the heating side the control current acts through electro-magnetic valves which open or close the steam passage.

Systems more or less inspired by the American types have been adopted in various other countries. On the *South-Manchuria Railway,* for instance, the « Asia » train, which has a steam-ejector system, and the equipment supplied to

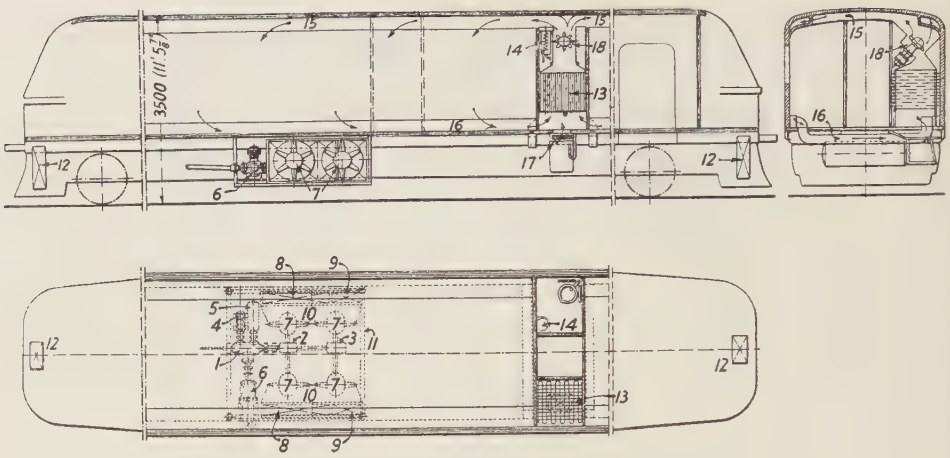


Fig. 6. — Lay-out of air-conditioning plant and accessory equipment on the diesel railcars of the Egyptian State Railways.

Legend:

1. Gearbox of accessory equipment.
2. Gearbox of fan No. 1.
3. Gearbox of fan No. 2.
4. Braking compressor.
5. Lighting dynamo.
6. Refrigerator compressor.
7. Cooling fans.
8. Radiator of diesel engine.
9. Freon condenser.
10. Casings of fans.
11. Frame for accessory equipment.
12. End condensers.
13. Evaporators.
14. Drinking-water cooler.
15. Air delivery duct.
16. Used-air suction duct.
17. Fresh-air intake.
18. Electric air-circulation fan.

the *Federated Malay States Railways* by Messrs. *Stone,* may be mentioned.

The Hungarian *Ganz Company* has constructed for the *Egyptian Railways* a number of railcars with a compression system of refrigeration using freon. In these cars the compressor is driven direct from the diesel traction unit, and the plant has a capacity of 14 000 to 16 000 frigories (55 500 to 63 500 B. T. U.) per hour. This arrangement was necessary owing to the lack of space, and its adoption enabled a very considerable economy in weight to be obtained (fig. 6).

European air-conditioning systems.

In France refrigerating equipment of the compressor type has not been proposed, except in the case of special colonial stock or high-speed railcars having the windows permanently closed in order to reduce air resistance.

Compressor equipment. — In the first category, mention may be made of the equipment designed by *Messrs. Brissonneau & Lotz* for vehicles destined for use in Indo-China, and in the second, the equipment adopted by the *Société Franco-Belge* for the high-speed railmotor trains under construction by them for the *Nord Railway.*

The *Brissonneau & Lotz* equipment differs from the American types by the cooling agent utilised, this being ammonia, and by the type of compressor.

Freon is not yet manufactured outside America, and is a fluid very difficult to store owing to its capacity for passing through the least porous of materials; it would therefore be difficult to maintain supplies in Indo-China,

whereas ammonia is easily available locally at low cost (10 times cheaper than freon). The use of ammonia necessitates the condenser being mounted below the underframe in order to guard against any risk of leakage into the vehicle. The evaporator has to be absolutely air-tight and is entirely welded. The compressor is coupled direct to a 7.5 h.p. motor and is designed for two stages of compression, the intake being obtained by rotary vanes giving a high degree of efficiency, and output being through ultra-light multiple valves.

The whole of the refrigerator and conditioning equipment is mounted together in a single chassis. The conditioning side includes a viscous filter, a cooling plant consisting of the evaporator of the refrigerator unit, a low-pressure steam heating installation and the circulating fan. The equipment is designed for an outside temperature down to + 4° C. (39° F.) in winter and 30° C. (86° F.) in summer, with a humidity factor up to 85 %, and within these limits will maintain an inside temperature of 15° to 18° C. (59 to 64.4° F.) in winter and 25° C. (77° F.) in summer, with 60 % humidity. In order to obtain the hygrometic correction the air temperature is first brought to its dew point of 15° C. (59° F.) which corresponds to 60 % humidity at 25° C. (77° F.). The air capacity is 1 800 m³ (63 600 cu. ft.) per hour, including fresh intake of 600 m³ (21 200 cu. ft.). Self starters are incorporated, with press-button control.

The rail motor trains constructed by the *Société franco-belge de Matériel de Chemins de fer* are three-coach units, the first and third being motor units. The centre coach, which contains the 1st-class accommodation, is equipped with a complete air-conditioning installation. Beneath the coach towards the middle are mounted two electric fans having a capacity of 1 300 m³ (45 900 cu. ft.) per hour, for blowing air each to one end of the vehicle through a duct leading to the air distributor running along the middle of the ceiling. Extending along the whole length of the distributor is a slot on either side through which the air is delivered tangentially to the ceiling, falling thence down the sides of the coach before diffusing. This method of distribution, due to *Mr. Moreau-Febvre*, is also found in the vehicles of the *French State Railways*. The vitiated air is withdrawn through outlets under the seats, and returns via ducts under the floor to the fans after being supplemented by fresh air drawn in from outside, and being filtered.

Heating takes place in the air flow from each of the fans, a hot-water system being employed, fed from a special boiler, and circulated by a pump.

On the intake side of the fans there is a cooling plant consisting of the evaporator of a « Glacia » machine. This machine has two drums mounted on a common hollow shaft through which the fluid enters and leaves. The drums consist of a series of finned copper tubes arranged in the form of a cylinder. The refrigerating fluid (freon) circulates through the tubes and is cooled by the air flowing from the inside to the outside of the cylinders, the flow being activated by the rotation of the drums.

One of the drums comprises the compressor which is of the two-cylinder V-type, the pistons being driven from the common shaft and prevented by a counterweight from being carried round with the rotating drum. The condenser consists of the drum itself.

The other drum forms the evaporator of the refrigerating cycle, and acts as the cooler for the circulating air.

This machine has the advantage of not incorporating any piping or stuffing box, and being therefore intrinsically tight. Further, the moving parts all run

in oil and require practically no attention.

The space available is not sufficient for these types of plant to have any very great capacity.

Before leaving the systems incorporating a compression refrigerator, reference must be made to an entirely original system evolved in Belgium by the *Westinghouse Brake Company*, and installed in Belgian and Dutch vehicles and in a streamlined train of the French *P. L. M. Railway*. This system is based on a « Lèbre » heat pump and the cooling agent is the air itself which, while not, of course, passing to the liquid state, undergoes a series of compressions and expansions that raise or lower its temperature. In winter the interior air bathes the heat exchanger, and the exterior air the cold exchanger. In summer, the opposite is obtained by inversion of a set of valves. The same apparatus, therefore, furnishes heat or cold at will. The component elements are as follows :

1. A compressor-expander with pressure exchange, requiring no other source of energy than that necessary for bringing the capacities into communication with each other and with the heat or cold exchanging zones;

2. A compensator, requiring only such energy as is necessary for compressing the expanded and heated air in order to compensate for the reduction in volume which it undergoes when passing through the cold source. The compression factor at this stage is about 0.25 to 0.30 of the total volume, including leaks and the compressor output;

3. Two fans propelling the compressed or expanded air;

4. Two heat exchangers, one of which gives up heat to the cold ambient air, whilst the other takes heat from the warm ambient air.

A detailed description of this equipment would take up too much space in an article of this kind, but it may be noted that at no point of the cycle does the pressure exceed 700 gr./cm² (10 lb. per sq. in.), that the maximum temperature range is in the neighbourhood of 50° C. (90° F.), and that there is no mechanical friction other than at the bearings of the pump rotor. The performance of such apparatus is excellent and the cost of upkeep extremely low.

Water evaporation systems. — Whereas the refrigerator systems require a relatively large amount of power, the methods we are about to describe are susceptible of economical application whilst giving quite adequate results for temperate climates.

The *Office central d'Etudes de Matériel de Chemins de fer* in collaboration with *Mr. Moreau-Febvre*, consulting engineer, and *Messrs. Nessi Frères*, of Montrouge, has designed equipment for the ventilation, heating and cooling of coaches in high-speed trains, the cooling capacity, whilst not approaching that of the American installations, being sufficient for our climates (figs. 7 and 8).

The equipment is all mounted under the vehicle and comprises :

(1) A chamber into which air is forced through oiled baffle filters, replaceable by other types if desired;

(2) A group consisting of :

(a) A heating system;

(b) A cooling system;

(c) A humidification chamber;

(d) A water separator;

(3) A fan for drawing air through the above-mentioned conditioning group and propelling it into the distribution ducts;

(4) The distribution ducts;

(5) The control gear.

The heating system consists of finned tubes traversed by steam reduced to a

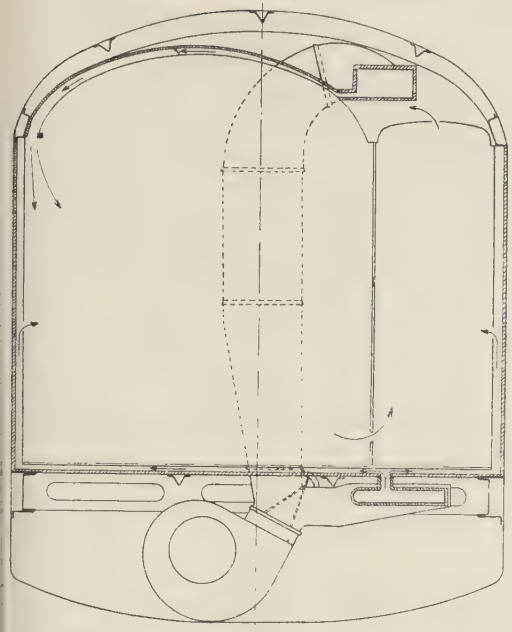


Fig. 7.

pressure of 1 kgr./cm² (14 lb. per sq. in.). The amount of steam admitted is regulated by a pneumatically operated valve controlled by a thermostat situated in the outflow of used air, and therefore in constant relation with the mean temperature of the vehicle, which is more or less the same in all compartments as the windows are not under the passengers' control.

The cooling system, of the « Amelior-air » type, consists of a series of very thin plates arranged in the form of a baffle, one side of each plate being in contact with the air flow proceeding to the compartments and the other being cooled by trickling water, pumped from a tank under the vehicle. The water is evaporated by a rapid current of air produced by an auxiliary fan and exhausted to the exterior.

The heated or cooled air passes into the humidification chamber, the process being effected by steam in winter and a

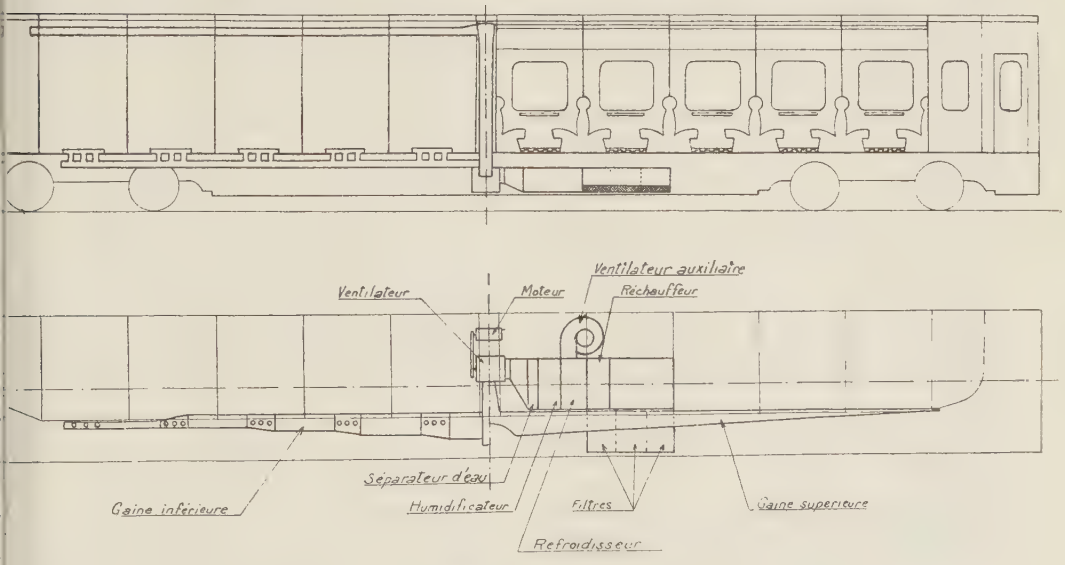


Fig. 8.

Explanation of French terms:

Filtres = filters. — Gaine inférieure = lower duct. — Gaine supérieure = upper duct. — Humidificateur = humidifier. — Moteur = motor. — Réchauffeur = heater. — Refroidisseur = cooler. — Séparateur d'eau = water separator. — Ventilateur = fan. — Ventilateur auxiliaire = auxiliary fan.

water spray in summer. Humidity is controlled by a hygrostat situated next to the thermostat.

After traversing the water separator, the air is picked up by the fan and propelled in two unequal currents into the two distribution circuits.

The first of these circuits consists of a duct running along the corridor underneath the floor, with a delivery branch opposite each compartment. Each branch is connected to a floor « board » consisting of fluted iron, the flutings running across the compartment and opening at each end into the space between the double walls of the vehicle, leading ultimately into the compartment or corridor through air inlets under the windows.

The second circuit is formed by a vertical duct between two compartment walls, opening into two horizontal branches above the corridor ceiling and having a continuous longitudinal slot through which the air passes into a cavity between the plywood ceiling of the compartments and a false ceiling of light alloy. This cavity increases in section from the cornice over the compartment door to the cornice over the window bay, where it opens into the compartment.

The used air of the compartments passes through outlets in the lower part of the doors and enters the corridor, which it leaves through a grill in the middle of the ceiling, thence to the exterior via the space between the two walls containing the ascending duct. The temperature and humidity controls are placed over the corridor ceiling in the air flow.

The plant is designed for an air capacity of 2 000 m³ (70 500 cu. ft.) per hour in winter, 1 500 m³ (53 000 cu. ft.) in the lower circuit, and 500 m³ (17 500 cu. ft.) in the upper. In summer the capacity is 4 800 m³ (170 000 cu. ft.) per hour, 1 200 m³ (42 500 cu. ft.) in the lower circuit, and 3 600 m³ (127 500 cu. ft.) in the upper. Water consumption is about 20 litres (4.4 Br. gall.) per hour.

As will be seen from these figures, the designers have endeavoured to obtain the sensation of summer freshness by considerably augmenting the air output and its speed of circulation rather than by cooling it to the extent that could be achieved by employing a cooling machine. The water-evaporation cooling unit is, in fact, only installed experimentally and is removable if found unnecessary. This would permit of considerable simplification and an appreciable saving in weight and cost.

The complete equipment as designed weighs about 1 ton and requires a motive force of 3 H.P.

Messrs. Neu, of Lille, have designed similar, though not so complete, equipment for a railcar, heating being obtained by circulation of the cooling water from the traction motor, and cooling by water spray in a chamber through which the ventilation air passes. It is claimed that a sufficient degree of cooling by water spray can be obtained without excessively raising the air humidity, this opinion being based on meteorological observations made in the Ile-de-France.

Systems without a source of cold. — In this last group are included the systems in which all cooling has been dispensed with, it being considered that in our climates abundant ventilation with air taken direct from the exterior is sufficient, even on hot days, and that in any case the number of exceptional days when air cooling would be desirable are so few as not to justify the installation of special equipment.

In England *Messrs. J. Stone & Co. Ltd.* offer an air-purifying and heating equipment (fig. 9), the control arrangements of which are particularly interesting. The used air is not recirculated. The exterior air passes first through a wool

filter and then through one of the viscosity type composed of a pile of metal rings covered with a film of oil. Part of this air then flows on to a steam heating unit and thence to the circulating fan. The remaining portion of the air passes direct from the filters to the fan. The proportion of warmed air is regulated by means of shutters operated by the control apparatus. The fan propels the air along the distribution ducts. The control apparatus includes a thermostat A actuated by the temperature of the exterior air, and a thermostat B exposed to the air flow leaving the fan. These two thermostats act on the two extremities of a lever, the centre C of which operates a rocking commutator controlling the direction of rotation of a small reversible motor M. This motor controls the inlet shutters of the heater and by-pass, and thus regulates the proportion of air heated. The arms of the levers CA and CB are varied in order to obtain the desired temperature in the compartment. A third thermostat influenced by the temperature of the heating unit by-passes the latter when not required.

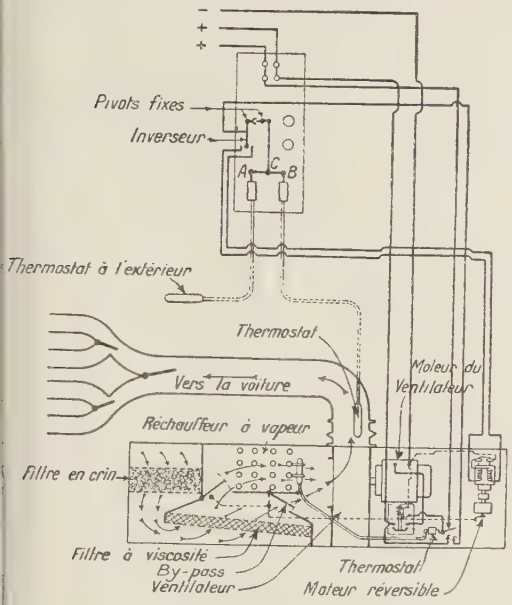


Fig. 9.

Explanation of French terms:

Filtre à viscosité = viscosity filter. — Filtre en crin = wool filter. — Inverseur = reverser. — Moteur du ventilateur = fan motor. — Moteur réversible = reversible motor. — Pivots fixes = fixed pivots. — Réchauffeur à vapeur = steam heater. — Thermostat à l'extérieur = outside thermostat. — Ventilateur = fan. — Vers la voiture = to the coach.

In France the *State* and *P. O.-Midi* Railways have perfected conditioning plant which is being installed on several series of coaches. These two railway systems are, or will be, partially electrified, hence the necessity to equip their stock with both steam and electric heating. As this dual radiator equipment is heavy, costly and bulky, there is every incentive to use air for heating purposes.

The *Moreau-Febvre* system of the *State Railways*, from which the *O. C. E. M.* system referred to above is partly derived, is based on the following principles :

1. Heating of the compartment walls so that, instead of being only a chilling factor as hitherto, they provide the major part of the heat required, leaving only the make-up to be supplied by convection;

2. Systematic ventilation;

3. Use of automatic regulators of great sensitivity.

In this system (fig. 10), the indrawn air is filtered and passed through a steam or electric heating unit whence it is propelled by a fan along two distributor ducts one of which is situated over the corridor and the other in the bottom of the side of the compartments.

Along the whole length of each compartment over the door is a small distributor chamber fed direct from the upper duct. A calibrated aperture forming the end of the chamber allows warm air to pass into the compartment at the rate of several metres per second, form-

ing a layer guided by the curvature of the roof. At window height the layer is 3 to 4 cm. (1 3/16 to 1 9/16 in.) thick. It is then narrowed by passage between the compartment side and a deflector plate and descends over the window bay and gradually diffuses.

The lower duct leads into pipes under the seats, from which the air is distributed at floor level.

A by-pass enables a proportion of cold air to flow into the upper duct so that

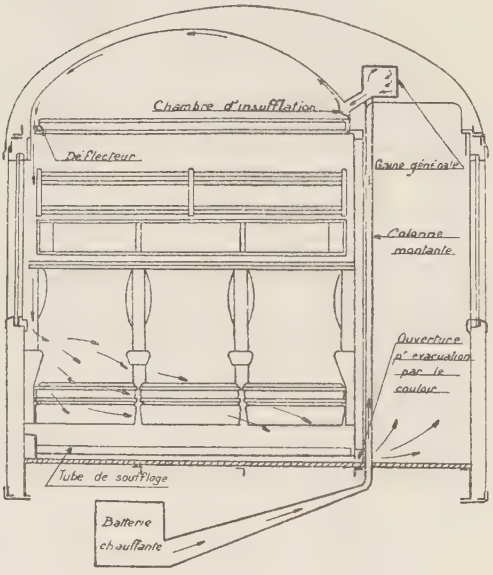


Fig. 10.

Explanation of French terms:

Batterie chauffante = heating unit. — Chambre d'insufflation = air distribution chamber. — Colonne montante = vertical duct. — Déflecteur = deflector. — Gaine générale = main duct. — Ouverture... = vent for evacuation of air to corridor. — Tube de soufflage = air distribution pipe.

its temperature is 10° C. (18° F.) less than that of the lower duct.

Used air is exhausted through the corridor through vents in the partition and is not recirculated.

The heating unit is fed through a thermostatic tube which reduces the steam to about atmospheric pressure so as to make the capacity of the unit independent of the position of the coach in the train.

Both steam and electric heating units are divided into three sections, one or two of which can be cut out of the control of the automatic regulators, thereby reducing the amplitude of the variations and giving a different range of temperatures. This is a distinct advance on the former « All or nothing » system.

The method of regulation is particularly interesting and will be made the subject of a detailed account in a future issue of the review *Electricité*.

In the system adopted by the *P. O.-Midi* the distribution of warm air is entirely at low level by means of inlets arranged along the walls.

Automatic regulation of temperature is obtained by means of two pneumatic thermostats placed in two check compartments and so adjusted that, when one is placed in the « too cold » position, the heater is switched in but can only be cut out when both the thermostats are placed in the « too hot » position.

The electric heating equipment of these coaches is particularly interesting in view of the fact that they are required to run on foreign lines operated at different voltages. The installation is therefore designed so that the heaters can be connected :

— In parallel for a.c. at 1 000 volts;

— In series parallel for d.c. at 1 500 volts;

— In series for d.c. at 3 000 volts.

The differential switching, which is entirely automatic, is done by special switch gear housed in a single chest mounted under the vehicle and comprising :

— Voltage selector relays which operate as soon as the intercoach heating cable is under tension, switch in the required combination of heating units,

and cut in appropriate fuses for the line-voltage;

— Connecting contactors controlled by the selector relays and fitted with automatic mechanical and electro-mechanical safety locks to prevent the heaters being switched on if for any reason the connections are incorrect or badly made;

— A main contactor controlling the heating installation, rated to carry any of the voltages referred to above and so designed that :

1. It cannot be closed if the heater connections are not correct for the line voltage;

2. It automatically cuts out if the connections are uncorrectly made;

3. When closed it mechanically locks all the differential contactors and switches the current off from their coils;

— An automatic starter for the fan motor controlled by the main contactor, so as to bring the fan into operation or stop it simultaneously with the heating system;

— Various other safety devices to protect the heating installation and the battery of accumulators supplying the control current.

The *P. O.-Midi* coaches can also be equipped with the ice cooling system referred to at the commencement of this article.

Conclusion.

I will be seen from the foregoing that the problem of complete air conditioning of railway vehicles appears to be solved in America, but that the methods employed would not be suitable for the European Railways. As regards the latter, the matter is still the subject of many discussions and experiments, and the French Railways appear to be on the point of evolving systems of great interest.

[621.392 (.44) & 621.134.1 (.44)]

Welded cylinders for 2-8-2 locomotives,

French National Railways Company,

by E. MONIER,

Chef divisionnaire d'atelier, at the Périgueux Workshops (*).

(*The Railway Gazette.*)

Many breakages have occurred in the fixing brackets of the cast-iron cylinders of locomotives of the former P. O.-Midi Railway, Classes 141.827-141.960. These are often incapable of being repaired by oxy-acetylene or electric welding, and therefore involve premature replacement. The engines are of American construction, and the breakages are due to fatigue resulting from a defect of design in the attachment. It was therefore decided to test the practicability of building the cylinders by welding together plate components, and in order to ascertain fully the possibilities and technique of this construction and the behaviour of the welded cylinders in service, the work was entrusted to the Périgueux workshops.

Special importance is attached to making the welded cylinders strictly interchangeable with the cast iron cylinders, and to retaining the same form and dimensions in the steam passages, so that no modification would be required in any part of the locomotives. This condition has been fulfilled, with the single exception of the steam supply pipe. A quarter-scale model was first built to settle the lines of welding, their accessibility, and the sequence of operations, taking into account the available facilities for machining at the Périgueux shops. Three sets of parts are built separately for each cylinder, *viz.* : (1) The steamchests and cylinders with their connecting passages; (2) box framing; (3) exhaust pipes.

The rings or barrels forming the steamchests and cylinders are welded by oxy-acetylene blowpipe, but all the other joints are arc-welded, using boiler welding electrodes in all cases, excepting the attachment of the end flanges of the cylinders, and the welding of the ribs on the latter and the fixing bracket. In these cases it is important to secure maximum strength, hence high quality forgeable electrodes are used.

Cylinder construction.

The cylinder consists of a ring of grade B steel plate 20 mm. (25/32 in.) in thickness, bevelled by an oxygen cutting torch and welded longitudinally by an oxy-acetylene burner. After being welded, the cylinder is turned extern-

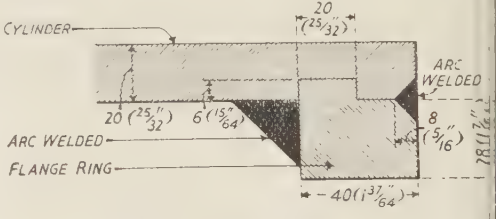


Fig. 1. — Showing arc-welded joints between cylinder and end ring.

ally at each end for a width of 50 mm. (approx. 2 in.) to receive two rings as shown in figure 3. As these rings carry the studs holding the cylinder covers, special care is needed in their welding.

(*) Reproduced from *L'Apprenti P.O.-Midi.*

The next step is to cut out the ports and weld on the four ribs of 30-mm. (1 3/16 in.) plate. Figure 2 shows one of the cylinders ready for assembly with the steamchest.

Fig. 2. — Main cylinder of welded plate construction ready for assembly with valve cylinder.

The latter is formed from a ring of 15-mm. (19/32 in.) B steel plate. After being welded longitudinally with an oxy-acetylene torch, it is bored and turned externally at each end prior to the attachment by welding of pieces as follows (see fig. 3) : Six rings, the interiors of which are roughed out to permit subsequent boring to the diameter of the two distributing sleeves, one at each end of the steamchest, and seven longitudinal distance pieces forming a stop for the sleeves; also two end rings for the studs holding the end covers. After welding on these attachments, the ports are cut and the steam supply pipe welded in position (fig. 4).

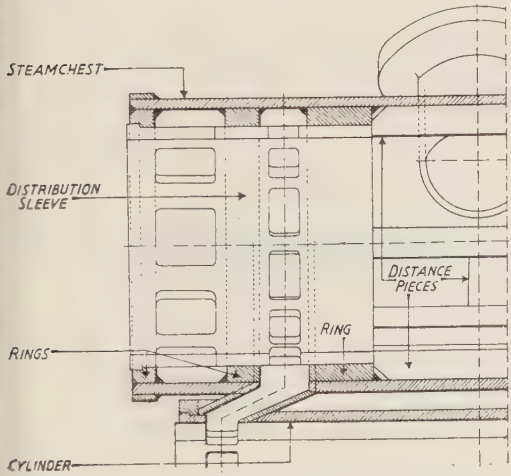


Fig. 3. — Showing construction of steamchest built from welded plate component.

Fig. 4. — Steamchest with ports cut and steam supply pipe welded on.

The steam connections between the steamchest and cylinder consist of 10-mm. (0.394 or slightly over 3/8 in.) plates formed as half shells and joined in pairs by arc welding the joints.

In order to ensure parallel setting and correct distance between the axes of the steamchest and cylinder, the two are mounted in a pair of hardwood jigs as shown in figure 5, and steel wedges are placed at *a* and *b* to hold the cylinders securely during the welding on of the

steam connections between the two-cylinders. After this operation the cylinders appear as in figure 6.

Box-frame cylinder supports.

The cylinder supporting frame is in box form and consists of five main parts,

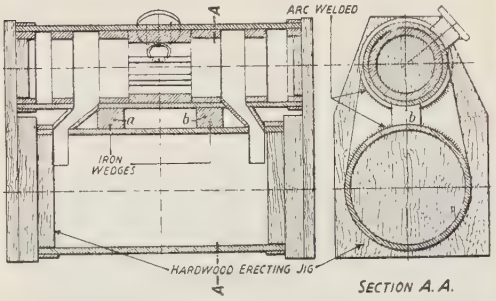


Fig. 5. — Cylinder and steamchest mounted in assembly jig, ready for the attachment of the steam connections.

viz. : the back plate of B steel, 30 mm. (1 3/16-in.) in thickness; the bottom plate of the same material and thickness; the fixing bracket of forged B steel; and the front and rear ends of 15-mm. (19/32-in.) plate. These pieces are assembled by electric welding, with the addition of numerous reinforcing gussets.

Most of the welds being between the bevelled edges of plates not in the same plane, it was recognised that points of contact must be provided between the plates at intervals (about every 300 mm. or 11 13/16-in.) in order to avoid, as far as possible, deformation due to contraction (fig. 7). After the main welds have cooled, the contact bosses are chipped out and the cavities filled by welding.

Before being welded to the bottom plate of the box frame, the fixing bracket is fully machined, except for the bearing surface for attachment to the main frame of the locomotive; this surface is rough machined, with an allowance for

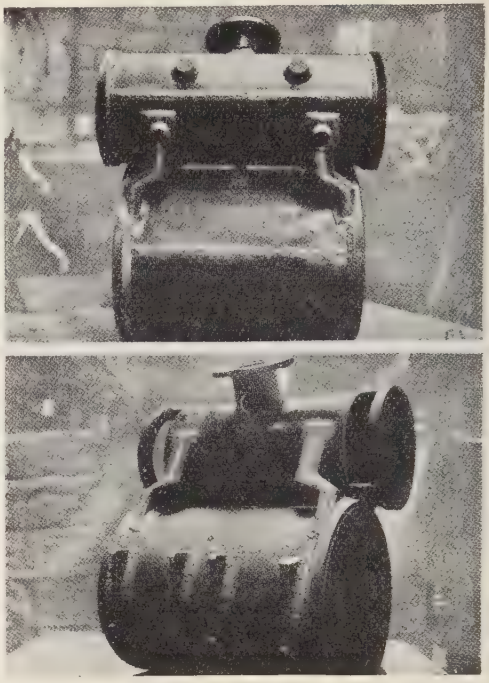


Fig. 6. — Assembled cylinders and steamchests of all-welded construction as seen from both sides.

finishing. The fillet welds of the fixing bracket on the lower wall are reinforced by five 60-mm. (nearly 3/8 in.) spot welds. Figure 8 shows the completed support.

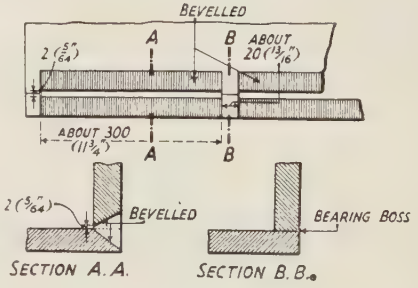


Fig. 7. — Temporary bearing bosses (subsequently replaced by welds) prevent deformation of angle-joints by shrinkage.

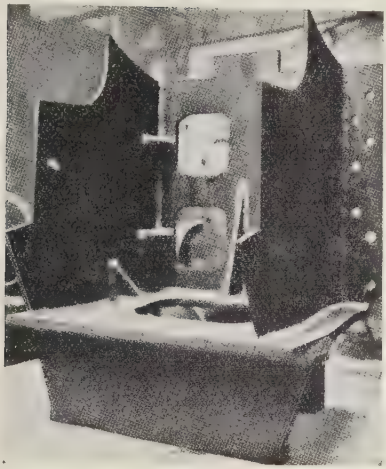


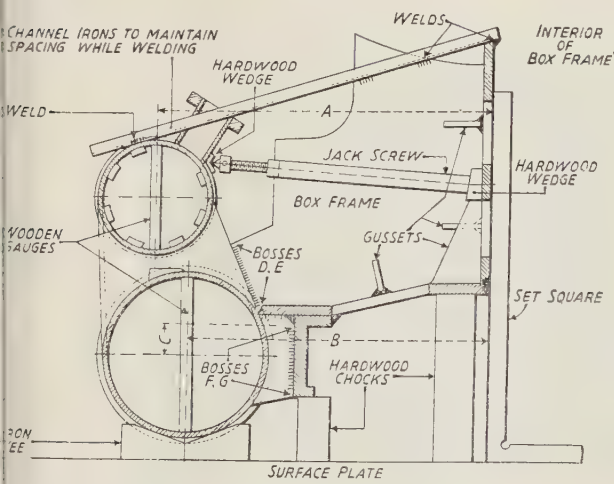
Fig. 8. — Welded box-frame support.

Fig. 9. — Exhaust steam pipe built by arc-welding sheet metal parts.

As the planers in the Périgueux shops are not capable of machining, with the cylinder in place, the face of the fixing bracket for the locomotive frame and the seating for the support of the front compensating beam on the bottom plate, the boxing had to be marked out and the parts mentioned planed before proceeding further.

Mounting cylinders and exhaust pipes.

The exhaust steam pipes, of Y-shape, are built up by arc-welding together six



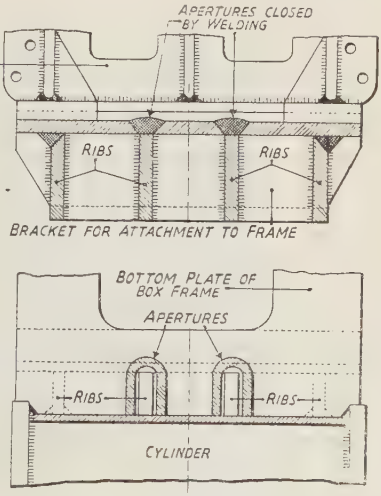


Fig. 10 (left). — Cylinder, steamchest, and supporting frame mounted ready for welding.
Fig. 11 (right). — Showing method of welding ribs to fixing bracket and lower plate of cylinder boxing.

pieces cut from 8-mm. (5/16 in.) sheet metal. A blade piece guides the steam flow at the junction of the two branches of the pipe. The completed pipe is illustrated in figure 9.

In mounting the cylinders in their supporting frame it is essential to establish accurately, in accordance with the drawings of the locomotive, the distances A, B (fig. 10) between the centre lines of the steamchest and cylinder, and the back plate of their supporting box-frame, allowing for 2 mm. (5/64 in.) contraction after welding; also, the vertical distance C between the centre line of the driving cylinder and the seating of the fixing bracket on the frame.

The arrangement used for this purpose is shown in figure 10. Bearing bosses

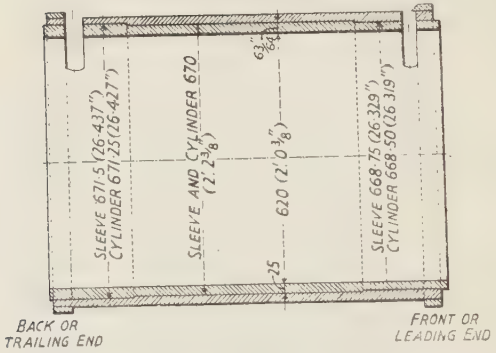


Fig. 12. — Reinforced and gusseted connection between end of cylinder boxing and steamchest.

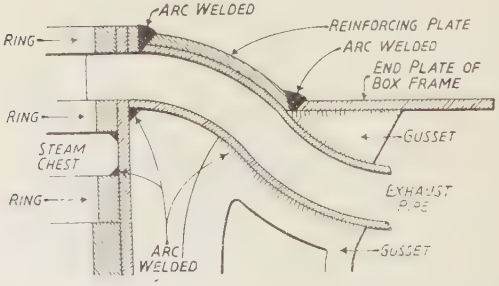


Fig. 13. — Details of cylinder liner.

forming points of contact between the parts are provided at D, E, F, G, to reduce contraction to a minimum, and the welds are made in the following order : bottom plate of box-frame to driving cylinder; front and rear ends of boxing to the steam pipes between steamchest and driving cylinder; four ribs on the fixing bracket; and four ribs on the bottom plate of the box-frame. In order to permit the welding of the two middle ribs, apertures are cut in the lower plate of the box-frame and closed by welding, as indicated in figure 11.

No difficulty is experienced in putting the exhaust pipes inside the cylinder boxing between the steamchest and the back plate, and then welding them to these two parts. Connection between the front and rear (leading and trailing) end plates of the boxing and the steamchest is established by a 10-mm. (about 3/8 in.) reinforcing plate, with gussets to consolidate the whole, as shown in figure 12. The top of the box framing is then closed by a 15-mm. (19/32 in.) plate, welded on to the steamchest and the end plates of the boxing.

Finally, the cylinders and steamchests are marked out, drilled and studded; bored by a Dyle and Bacalan portable machine; and fitted with their cast-iron liners. The drawing, figure 13, shows details of the cylinder liner, which is inserted from the trailing end. A hydraulic pressure test is now applied, 4 kgr. per sq. cm. (56.9 lb. per sq. in.) for the exhaust pipes and 20 kgr. per sq. cm. (284.5 lb. per sq. in.) for the combination of steamchest and cylinder with the exhaust pipes shut off, these not being built to withstand such a pressure. The completed construction is shown in figure 14, studded and tested, the plate forming the cradle for the reception of the boiler being laid on top of the box frame.

As there is no boring machine in the Périgueux shops capable of machining the 25-mm. (1 in.) plate forming the

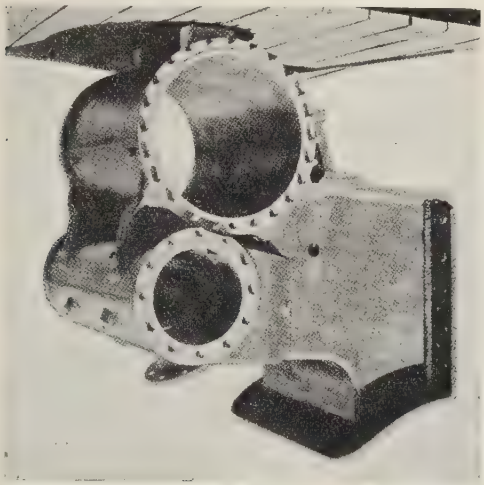


Fig. 14. — Completed cylinder and steamchest, studded and tested hydraulically.

smokebox cradle, after welding it to the cylinder box-frame, the following procedure is adopted : The group of cylinders is mounted on the locomotive frame, and the boiler, with the bottom of the smoke-box replaced, is mounted on the frame and adjusted accurately to height. The cradle plate, drilled and shaped approximately to the radius of the smoke-box, is slid between the cylinder box-frame and the boiler and fitted exactly to the smokebox ring by means of bolts and the application of a heating torch. The cradle plate is then tack-welded to the cylinder box-frame. At the same time, the opportunity is taken of levelling and adjusting the seating of the blast-pipe, which is welded to the cradle plate, the exhaust pipes and the back plate of the cylinder boxing. The boiler is then removed, the cylinders are detached, and the welding of the cradle plate on the cylinder boxing is completed.

Service trials.

A set of welded cylinders built as described above was fitted to locomotive 141.918 of the former P.O.-Midi Railways without any difficulty, and the regulation trials of the locomotive were completed without incident on August 24, 1935. In order to test the stability and smooth running of the cylinders under load, a traction test was then carried out with a train of five locomotives coupled together, when slipping of the wheels was caused purposely. This trial,

Fig. 15. — Welded cylinders, cylinder boxing, and smokebox carrier in position.

Fig. 16. — Locomotive No. 141.918, P. O.-Midi Railways, with welded cylinders.

also, was passed successfully, and locomotive No. 141.918 has since been in regular service from the Angoulème depot without any trouble arising from the cylinders. Up to May 24th, 1938, the total run of this locomotive provided with welded cylinders is 107 809 km.

So far as the author is aware, this is the first set of cylinders built by welding in France. Special expenditure was required on jigs and other equipment, and on a certain amount of experimental work in developing the technique. Nevertheless, the cost of the finished cylinders was less than that of cast steel cylinders, and no doubt a yet lower cost could be reached in regular production, intermediate between the cost of cast steel and cast-iron cylinders.

[625 154]

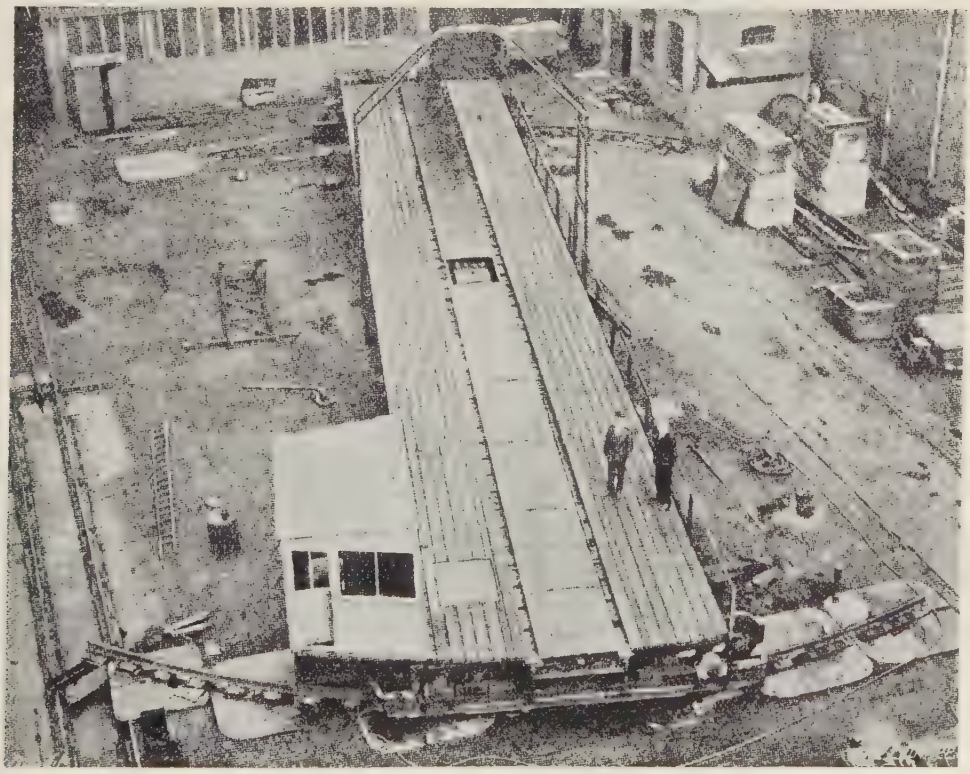
A 30-metre locomotive turntable.

(*The Engineer.*)

During recent years the increasing weight and length of locomotives and the economic necessity of rapid turning round has revealed certain disadvantages in the centre balance type of turntable. In order to obtain balance this type requires a considerably larger diameter than the wheel base of the engine accommodated. To get this balance before turning, time is lost in inching the engine backwards and forwards, and further time is lost in blocking up the ends for the engine to run on and off the table.

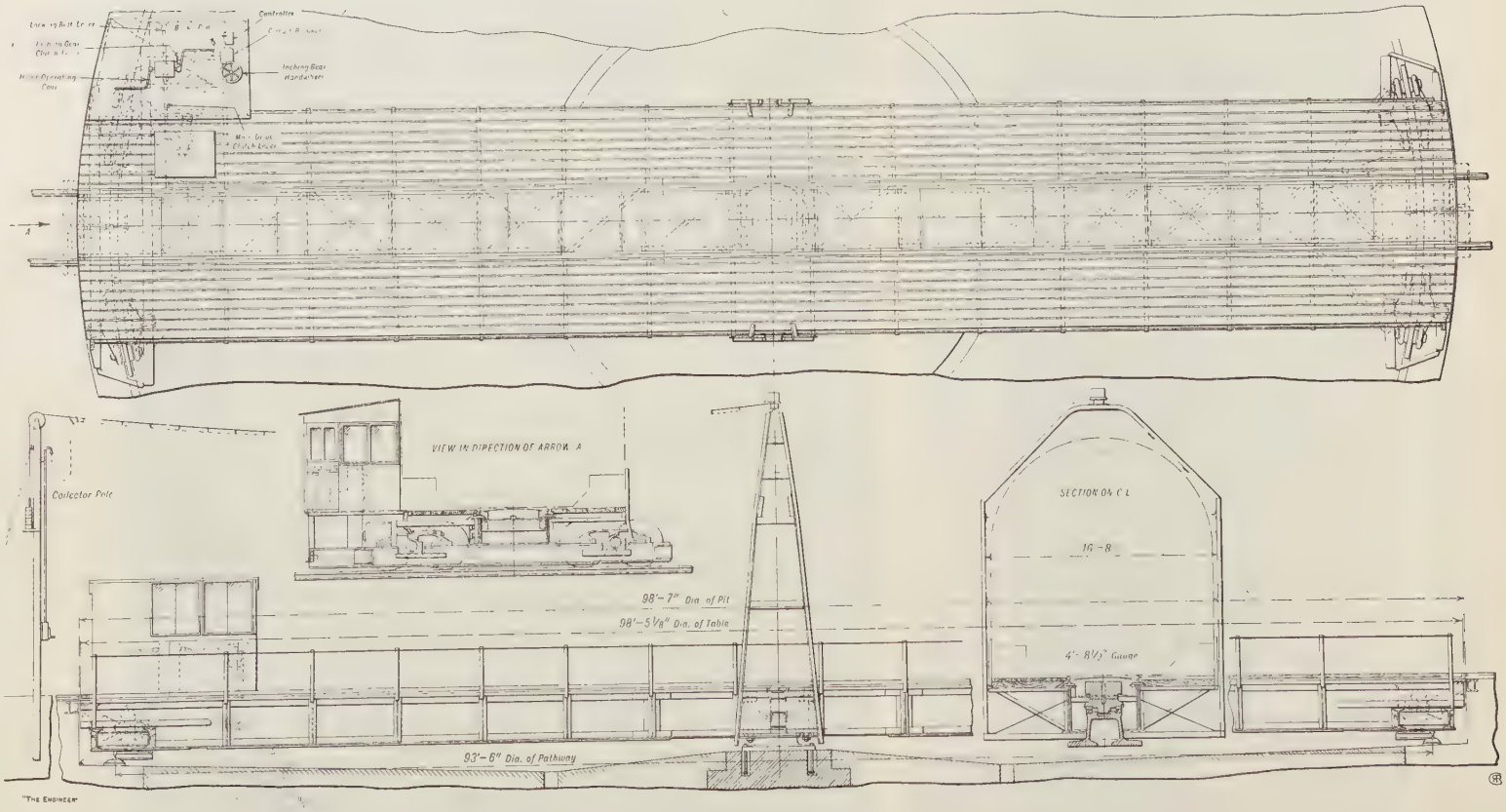
These disadvantages have been overcome by the introduction of the three-point support turntables, a number of which have been installed in Great Britain and abroad. In this design the load is distributed between the centre pivot and two sets of outer track wheels. This distribution of the load enables the tables to handle locomotives with wheel bases within a few inches of their dia-

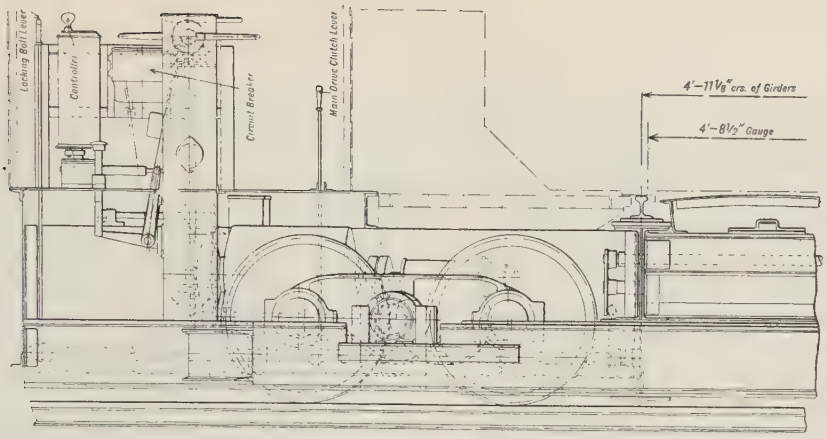
70-foot Rapier-Mundt turntable at York.

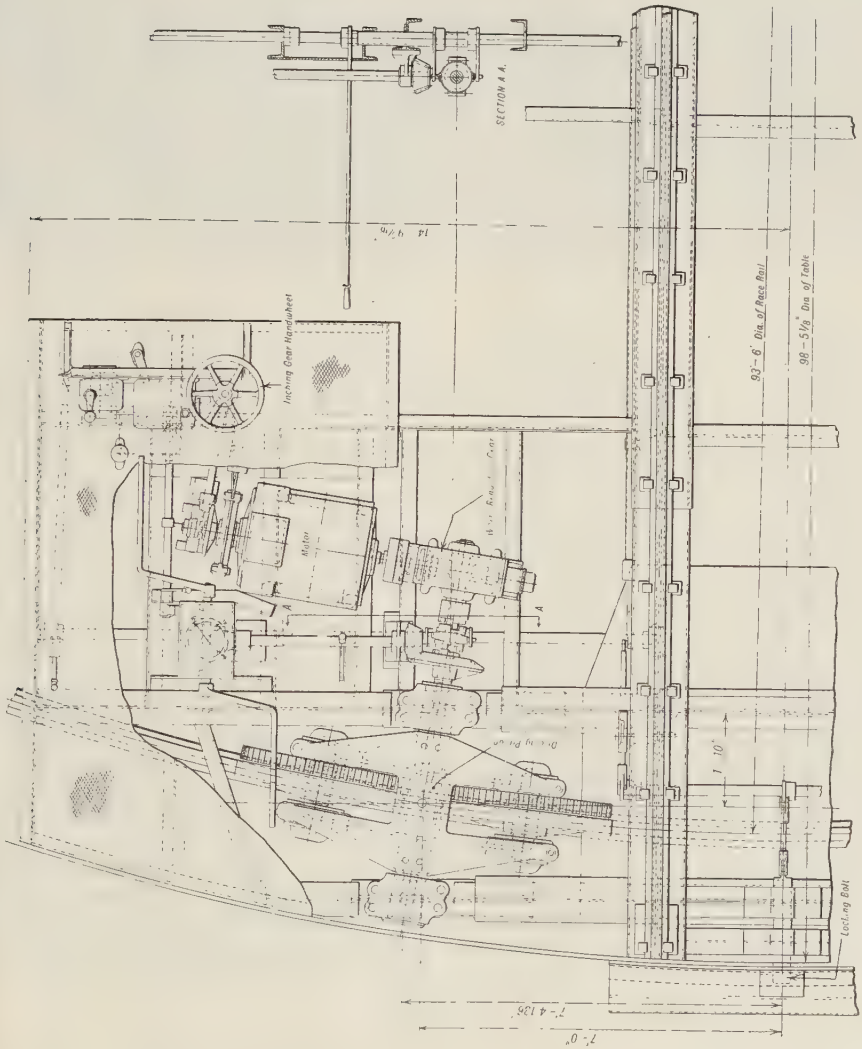


30-metre turntable assembled for test.

A 30-metre Rapier-Mundt locomotive turntable.




Locking Bolt Lever
Controller
Circuit Breaker
Main Drive Clutch Lever
4'-11 7/8" ers. of Girders
4'-8 1/2" Gauge


SECTION A A
Inching Gear Handwheel
Motor
93'-6" Dia. of Race Rail
98-5 7/8" Dia. of Cable
Locking Bolt
7'-4 1/2"
7'-0"

meter, regardless of balance. They are turned by either a hand or a power gear, which drives one of the track wheels. In order to obtain sufficient adhesion at the driving end for all conditions of loading, the main girders are usually articulated or otherwise flexibly jointed at the centre.

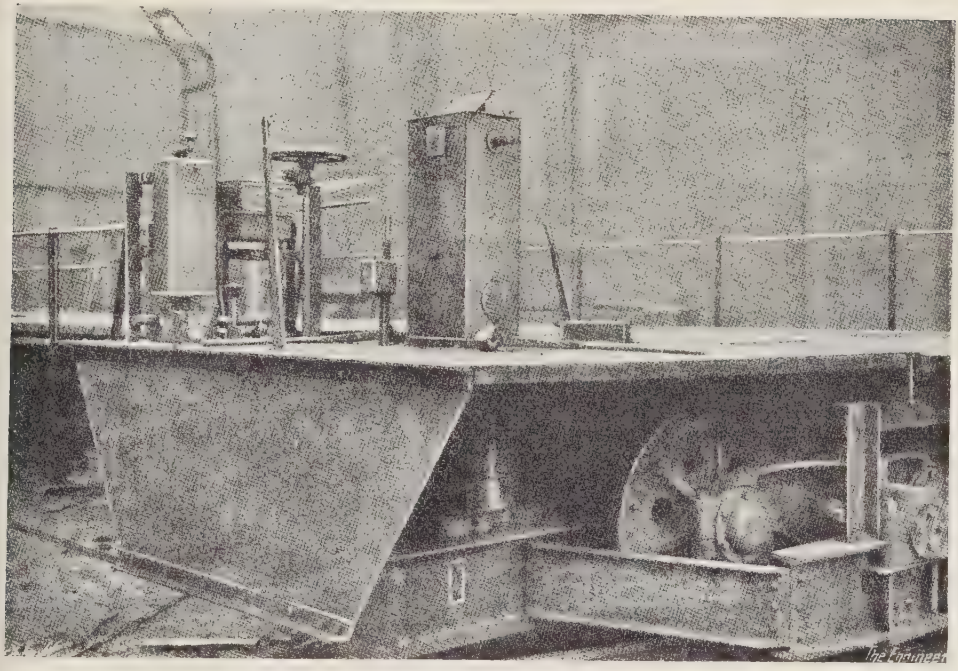
Some years ago, Mr. Mundt, an engineer of the Dutch State Railways, introduced a three-point support table which had continuous main girders. These girders are so designed that when the table is loaded at one end only sufficient adhesion load for turning is imparted to the track wheels at the other end. In this form of construction, when the table is under a balanced load, five-eights of the weight is supported by the central pivot and the remainder is evenly distributed between the two outer sets of track wheels. With an articulated three-point support table similarly loaded half the weight is carried by the central pivot and one-quarter by each of the two sets of track wheels. The distribution of the load on the Mundt type of table makes is easy to turn and little wear takes place on the outer track wheels and rails. The main girders are continuous beams, but the desired distribution of load between the centre and ends is obtained by varying the section of the girders so that the deflections and points of contra-flexure are such as to ensure the proper loads being on all three points of support under all loading conditions.

The fact that the Mundt turntable always has a load on all three points of support eliminates any impact on the table when a locomotive runs on or off. Furthermore, it does not require such a deep pit as the centre balanced type and the distribution of the load enables the use of much smaller foundations.

The sole manufacturing and selling rights of the Mundt turntable, both in Great Britain and many other countries have been acquired by Ransomes and Rapier, Ltd., of Ipswich. We reproduce a drawing of one of eight 30-m. Rapier-Mundt tables supplied to the Chinese Government, one of which is also illustrated herewith.

These tables are set in pits 30.5 m. in diameter and have a depth of 4 ft. 1 5/8 in. from rail to rail at the side and 4 ft. 10 1/8 in. from rail to base of the centre column in the centre. Designed for a working load equivalent to 330 tons distributed, they have been tested with a distributed load of 412 tons. Each consists of two continuous main girders, rigidly braced and supported on a centre pivot and a cross girder at each end. Each of the cross girders is carried on two articulated bogies, each of which runs on two rollers. The table is turned by driving both wheels in one of the bogies. The two main girders are of the single web plate type, 3 ft. 3 in. deep, with extra reinforcing flange plates in the bays between the centre and either end. The girders are secured together by full-depth plate diaphragms and the whole given transverse rigidity by horizontal angle bracing at top and bottom between the flanges of the girders. Cross girders are built between the main girders at the centre to take the suspension bolts from the centre cap. The main girders are cut down and strongly reinforced at each end to take the cross girders, on which are the pivots for the outer wheel bogies. Cantilever angle brackets are fitted on each side of the main girders at the diaphragms and carry 3 in. thick timber platforms. 5 ft. 3 in. wide. Between the rails is fitted a chequer plate platform with cambered plate to ensure proper drainage.

At each end of the table a rectangular section locking bolt is arranged on the centre line of the table to engage with pockets in the side of the pit at each running-on position. A single lever near the driving gear controls the locking bolts. Built-up steel girders are built into the pit side at each running on position and on them are mounted the end chairs of the leading-on rails. The



Driving bogie and control gear for locomotive turntable.

pockets for the locking bolts are built into these steel girders.

The centre about which the table pivots is supported on a cast steel column and consists of a double row ball thrust race housed in a heavy steel retainer on which the main steel cap is mounted. The cap is articulated to ensure even distribution of load on the ball race and is connected to the main structure by eight suspension bolts which allow centre height adjustments to be made to correct settling or other foundation irregularities. The ball race is protected against the ingress of dust or water and has pressure grease lubrication. Each of the four bogies pivots on a cast steel bearing bolted to the cross girders and set in a position radial to the centre of the table. The bogies are single-piece cast steel frames with axle bearings of the self-aligning double-row ball type. The 3 ft. 4 in. outer track wheels are cast steel centres with shrunk-on tires of chrome steel and into them are pressed high-carbon steel axles. On each wheel of the driving bogie a steel bevel wheel is spigoted and bolted. This bevel wheel is driven by a bevel pinion keyed to the driving shaft which passes through and is carried in the hollow trunnions of the bogie casting. The face angle of the driving bevels is less than 1 deg. and to ensure smooth running the teeth have been accurately generated from solid blanks. The bevel pinion shaft also carries a bevel drive for the hand gear and it extends to the worm gear-box for the electric drive. A clutch incorporated between the bevel and the worm gear-box enables the hand or electric drive to be engaged.

The hand gear drive is transmitted by a shaft and bevel gears to a winch standard with two removable crank handles. We are informed that when the table was being tested four men turned it with a load of 412 tons without any undue

exertion. The specified turning speed by hand is half a revolution in 4 1/2 minutes with the table carrying a working load.

Electric operation is effected by a 20-H.P. motor acting through an overdriven worm box to the bevel pinion shaft. With the electric drive in use the turntable makes half a revolution in one minute.

An extension to the rear end of the driving motor shaft is fitted with a brake — which is operated by a pedal in the cabin — and the electric inching gear. As may be seen in our drawing, this inching gear comprises a bevel wheel and a clutch which is geared through a vertical shaft to the hand control wheel. Using this inching gear in conjunction with the electric drive, one man can complete the final lining up of the tracks before setting the table locking bolts.

The controls are grouped together in a sheet steel cabin and comprise the crank handle standard, selector clutch lever for hand or electric drive, locking bolt lever, controller, circuit breaker, inching gear hand wheel, and clutch lever. In order to reduce the risk of damage likely to be caused by careless handling, a series of automatic interlocks has been arranged in a locking box to which the controller spindle and various levers are led. As a result of these interlocks the motor cannot be started until the locking bolts have been withdrawn and the locking bolts cannot be shot home whilst the motor is running; the motor cannot be started when the inching gear clutch is engaged and the clutch cannot be engaged whilst the motor is running; and the foot brake cannot be applied until the controller is at neutral. The change over from electric to hand drive can be made with the table in any position.

The motor is a 20-H.P. at 580-r.p.m. totally enclosed slip-ring induction reversing, one-hour rated unit, built for a 380-volt, three-phase, 50-cycles supply. Its windings have been specially impregnated for working in China.

The control gear, shown above, consists of a totally enclosed vertical drum type air break reversing controller with seven notches in each direction and auxiliary contacts for interlocking with circuit breaker in « off » position. An enclosed iron-clad, free handle, oil immersed, triple-pole circuit breaker with overload and no-volt releases is fitted. A separate protected type resistance is fitted in the house near the controller. The current is taken on to the table by an overhead cable connected to an enclosed slip-ring gear mounted over the centre of the table on a steel arch built up from the table and around the loading gauge clearance line. A steel pole with back stay is mounted at the side of the pit and a steel wire rope to which a cable is attached by insulated suspenders is stretched between the pole and the collector arch on the table. Even tension is maintained by weighing the end of the wire rope which passes over a pulley at the top of the steel pole. An ironclad triple-pole main isolating switch is fitted on the pole.

The outer track rails consists of twelve lengths of 100 lb. per yard flat-bottom rail, accurately curved to make up a complete circle of 93 ft. 6 in. diameter with proper expansion gaps and diagonally cut joints to avoid impact from the wheels. The rails, which are machined on the head and fish-plated together, are mounted in the pit on cast iron chairs to which they are clipped. The chairs are fixed down to the concrete foundation, care being taken to ensure a level setting of the complete rail ring before grouting up.

As previously stated a number of Mundt type turn-tables are now in service on the railways of Great Britain and we illustrate one of the 70-ft. diameter, electrically operated type, which has been installed by the London and North-Eastern Railway Company at York.

[621. 132 3 (.73) & 625. 232(.73)]

The Reading Company installs a light streamline steam train.

(Railway Mechanical Engineer.)

On December 13, the Reading Company placed in service on the Reading-Jersey-Central line between New York and Philadelphia, Pa., a lightweight streamline stainless-steel train of five cars built by the Edward G. Budd Manufacturing Co. The train is drawn by a *Pacific* type steam locomotive remodelled and streamlined at the Reading shops of the railroad. The lines of the locomotive conform with the sheathing of the cars, thereby presenting a continuity of line and surface throughout the train.

The train is made up of four chair cars and one tavern-dining car. The two end chair cars are constructed as observation cars to obviate the necessity of turning the entire train around after its arrival at either terminal. The end cars each contain 56 coach seats and 14 individual lounge chairs and the other chair cars each have 56 coach seats and 12 individual lounge chairs. The dining car accommodates 24 passengers at tables in the dining room and 27 passengers in the tavern or cocktail lounge.

The car structures.

All car sheathing and structure, except the end underframes, are the well-known 18-8 stainless steel. This is used in two strengths. The high-tensile material has a tensile strength of 150 000 lb. per sq. in. and is used where light weight and strength are of prime importance, and the low-tensile material has a tensile strength of 100 000 lb. per sq. in. and is used where workability, or special finishes, is required. The Budd Shotweld process was used in the fabrication of these cars.

Fundamentally, the structure of the cars is a modified girder wherein the roof and under structure serve as compression and tension chords, and the two modified Pratt-truss side frames are the webs. The end structure is designed to withstand buffing, traction, vertical and

lateral loads that are to be expected in service and as required by railway mail service. The end underframes are made of Cromansil steel, arc welded into a unit subsequently stress-relieved and riveted to reinforced body members.

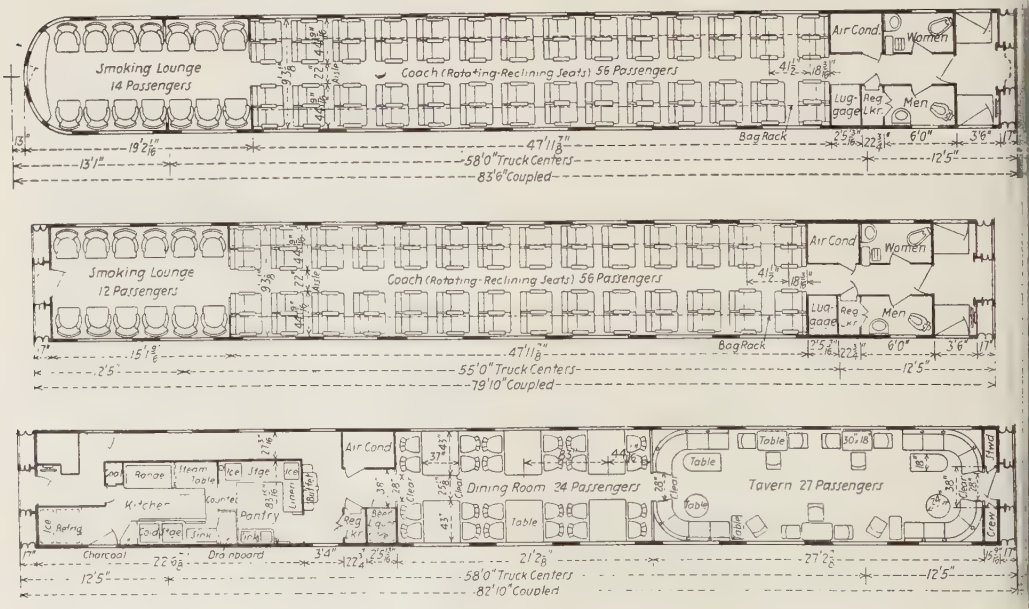
Doors are of all-metal construction, with a light of 1/4-in. shatterproof glass in the upper portion. The stainless steel used in the outside doors is in harmony with the exterior metal. The passenger steps are so constructed that they are raised into the car body when the vestibule trap doors are closed, presenting a smooth exterior appearance.

All passenger side windows are double-glazed, with 1/4-in. polished plate glass in the outer frame and 1/4-in. shatterproof plate glass in the inner frame, with about 1/4-in. air space between them The outer window frames are attached directly to the car body, while the inner sash is hinged at the top to facilitate cleaning. The window frames are of all-metal construction and the glass is held in place by molded rubber strips. The toilet-room windows are single-glazed, with shatterproof flashed opal glass.

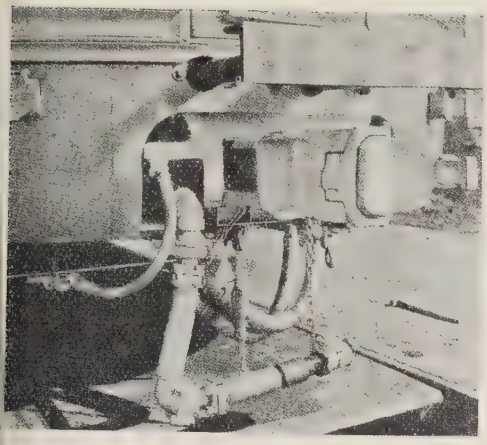
The cars are thoroughly insulated. The side walls, floors, roofs and ends are insulated with Stonefelt and the windows are of double-glazed construction to further reduce heat losses. The side, end and roof insulation is held in place by metal clips and bands. The sub-floor insulation is supported by flat sheets of stainless steel on the under side.

Air conditioning and lighting.

The Airtemp electro-mechanical air-conditioning equipment comprises a compressor-condenser unit of about 6 1/2 tons capacity mounted beneath each car floor, and an evaporator unit placed in a locker in each car body. A controllable amount of filtered fresh air is taken through louvered openings in the car roof, mixed with air drawn from the passenger compartment and forced through the conditioning unit and into



Floor plans of the cars in the new Reading lightweight streamline train.



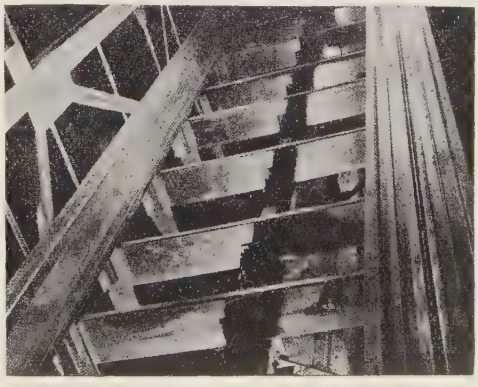
One of the controlled slack couplers.

he distributing duct which is made of galvanized steel insulated with 1/2-in. Deadnox. This duct, which is placed above the ceiling between the light ducts, allows the air to escape through a concealed narrow slot in the lower edge of each side. Incorporated in the evaporator is a steam heater coil of about 30 000 B. T. U. capacity for use when heated air is required. Finned copper-tube steam radiators are placed at the base of the walls for general car-heating purposes. The Vapor heating system is employed, with Barco steam-heat connections between the cars.

This train is the first complete installation of the Vapor correlative control of both heating and cooling. With this control system the only manual operation required is in changing over from cooling to heating or vice versa. In cooling the differential between the inside and outside temperatures is effected automatically under the control of an adjuster thermostat located in the fresh-air intake. With an outside temperature of 100° or higher the inside temperature in the car is under control of an 80° maximum thermostat. For each 3° drop in outside temperature below 100° F. a drop of 1° is automatically effected in the temperature inside the car until the inside temperature has dropped to 72°. Should the outside temperature drop below 72° with the control set for cooling, the adjuster thermostat is automatically dropped out of the circuit, thus preventing any further lowering of the inside temperature under outside temperature control. Both the overhead and floor heat are tied in the same automatic circuit controlled by an adjuster thermostat, in this case located in the recirculating air duct. This is set for a 3° differential between the overhead and floor heat.

A separate ventilation system is installed in the kitchen. Fresh filtered air is delivered directly to the kitchen without passing over evaporator coils and is moved by exhaust ventilators, the capacity of which slightly exceeds that of the intake blowers. One intake blower feeds two ducts which extend along the sides of the doorway between the kitchen and dining room. These ducts deliver a curtain of air across the doorway, which is drawn into the kitchen by the action of the exhaust fans, thereby preventing the escape of kitchen odors.

The electrical system operates at 32 volts d. c. The current is supplied by



Details of the underframe and side construction.

Exide 1 000-amp./hr. batteries and the Safety Car Heating & Lighting Co. 20-kw. generators. The batteries are carried in a stainless-steel battery box mounted beneath the car. The generator is mounted longitudinally under the car and connected to an axle by the D. R. drive — a combination of vee belts, bevel gears and longitudinal drive shaft.

Provision is made for the operation of the air-conditioning equipment from a standby source of power. The motors are built for operation on both 32-volt d.c. and 220-volt a.c. The cars are equipped with both d.c. and three-phase a.c. receptacles for receiving standby power.

Direct and indirect concealed lighting is used in various parts of the cars. The indirect lights are concealed in two ducts recessed in the ceiling. The ducts are painted white and reflect light onto the car ceiling to create an almost shadowless illumination in the car. Direct lighting supplements the indirect system in the lounges, dining room and tavern. The cove over the windows is constructed behind the side ceiling lining and is glazed on the under side in a horizontal plane, the light from this source offering comfortable reading conditions to passengers in the sections so equipped. Bag-rack fixtures in the coaches are centered over each seat and concentrate their light downward.

Weights of cars in Reading lightweight streamline train.

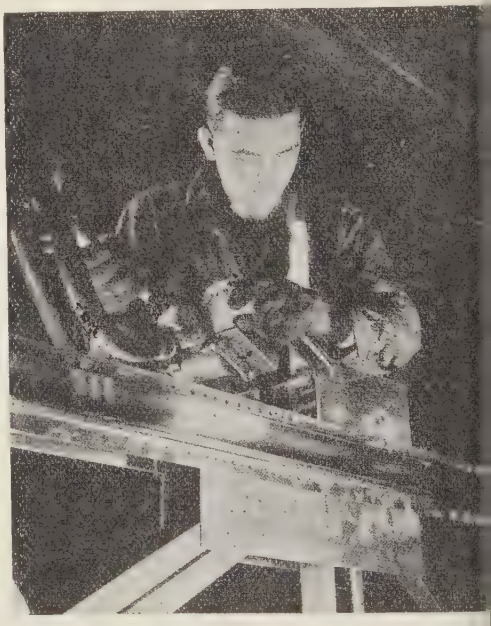
Car No.	Weight, lb.
1	98 914
2	95 774
3	110 399
4	96 024
5	98 914
Total	500 025

The vestibules, aisles and toilets are lighted from overhead fixtures recessed into the ceilings of these sections.

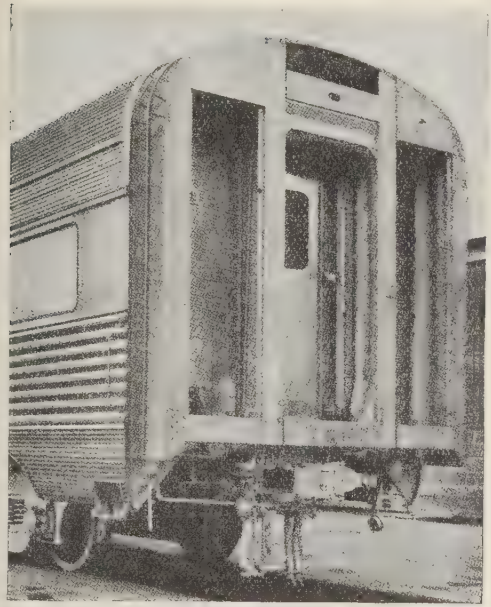
The power brake is the Westinghouse Air Brake Company's electro-pneumatic modified H. S. C., operated by air pressure supplied by the locomotive, and controlled electrically. There is a belt-driven speed governor on each of two trucks on the train, which controls braking pressures at various speeds. The hand brakes are of the wheel type mounted on a lightweight housing.

The trucks.

The cars are carried on four-wheel trucks of the conventional equalized swing bolster type. The wheel base measures 9 ft. Each truck is fitted with four Houde hydraulic shock absorbers to dampen the lateral swing action. Vibration and sound deadening is controlled by the use of insulation at strategic points. All castings are alloy steel



A side frame in process of fabrication.



End of coach showing the full-width diaphragm.

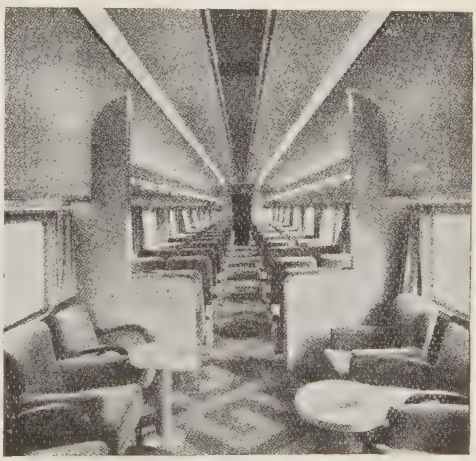
heat treated, and the equalizers are forged steel, machined all over. The axles are fitted with 36-in. multi-wear wheels and Fafnir combined ball- and roller-bearing journal boxes. The radial load is carried on hardened solid rolls, while the lateral thrusts are resisted by high-capacity ball bearings. The bearing housing is of alloy steel, hardened, and is reversible for double wear. The trucks are fitted with the Simplex Unit Cylinder clasp brakes.

Interior treatment.

The end cars are identical in layout and structure, but differ in color treatment. In one of the cars the main features of the colour scheme are olive brown walls and coral upholstery; in the other they are aquamarine and chamois. Similarly, the second and fourth cars are alike in arrangement, but one is decorated with clay color on the side walls and green upholstery, and the other with olive-green walls and brown frieze upholtery. In addition to the roller shades, drapes are placed at the sides of the windows in all of the cars.

The floor plans indicate that each of four cars has a smoking lounge. Those at the ends of the train each seat 14 passengers in individual chairs and each is fitted with a table in the rounded end of the lounge, with shelves at the sides. The lounge in each of the other two coaches seats 12 passengers in individual chairs. The passenger compartments in each of these four cars is fitted with the rotating type, two-passenger seats, with individual reclining backs and sliding cushions. The cushions are molded sponge rubber and the backs are of spring construction. The individual arm chairs in the lounges are of wood-frame construction with molded sponge-rubber cushions and spring backs. The flooring in all four coaches in Linotile in a checkerboard pattern, the colors varying from car to car. Each car has a man's and a woman's toilet adjoining the vestibule.

The middle car in the train is a 24-



Looking from the lounge section in one of the coaches.

passenger dining room, with a tavern compartment seating 27 persons at wall seats and individual chairs at small tables, and a kitchen and pantry. The kitchen equipment comprises a refrigerator at the extreme end of the kitchen; a coal range and water boiler; a broiler; two coal bins, one with a crew's locker above it; a steam table; a coffee urn, which stands on the steam table but with the faucet projecting into the pantry; two sinks with swivel faucets, one in the kitchen and one in the pantry; a service shelf; a low kitchen refrigerator; a fruit chill box; a liquor refrigerator and liquor locker, and numerous small lockers and compartments for the storage of dishes and supplies. All equipment in the kitchen is made of stainless steel, with the exception of the coal-burning range, and all exposed side wall is covered with sheet stainless steel. The miscellaneous hardware is chromium plated.

The dining room and tavern are decorated in olive brown with touches of red and the upholstering is tan leather with red piping. The floor covering in both rooms is Seamloc carpet in raisin and henna. A Velvetex pad is placed under the carpet.

The locomotives.

The locomotives, two of which have been streamlined for use with this train, are the road's Class GISA *Pacific* type, some of which were originally built by the railroad at its Reading shops. The two locomotives in question are from this group. They develop a tractive force of 40 900 lb., with a boiler pressure of 220 lb., cylinders 25 in. in diameter by 28 in. stroke, and driving wheels 80 in. in diameter. The boiler has a total evaporative heating surface of 2 984 sq. ft. The superheating surface is 621 sq. ft. and, originally built for anthracite service, the firebox has a grate area of 94.5 sq. ft.

The vestibule open, showing the step operating mechanism.

The locomotive and tender have been shrouded and streamlined with stainless and painted steel, the latter conforming to the lines of the coaches. An appearance of continuity of the sides of the coaches and tender has been effected by extending the sides of the latter beyond the tender end sill to enclose the rounded observation end of the forward car. A horizontal extension from the top of the tank produces a similar effect in the roof lines by enclosing the hooded portion over the end of the car. Louvers in the front of the locomotive shrouding near the top admit air which flows out of a port in the top of the shrouding just back of the stack to function as a smoke lifter. The estimated weight of the engine and tender is approximately 460 000 lb.

[385. (08. (73)]

Annual report of the Interstate Commerce Commission.

(November 1, 1936, to October 31, 1937.)

(Railway Age.)

The fifty-first annual report of the Interstate Commerce Commission, submitted to Congress on January 3, is a 177-page document covering in the usual form the regulatory body's activities during the period from November 1, 1936, to October 31, 1937, and setting forth general observations on conditions in the transport field. As in past years, several legislative recommendations are included, some of them carried forward from previous reports.

Legislative recommendations.

After leaving for later special consideration its views with respect to Motor Carrier Act amendments and regulation of freight forwarders, the commission recommends :

1. Enactment of a new law for comprehensive regulation of interstate air transportation.

2. That its previous recommendations looking to the regulation of common and contract water carriers be followed.

3. That the foregoing include appropriate legislation for the regulation of wharfingers.

4. That non-carrier railroad subsidiaries be brought within I. C. C. jurisdiction.

5. That the fourth section be amended by eliminating the so-called equi-distant clause, but not otherwise.

6. That section 15 (1) and (3) of Part I be amended « to enable us to prescribe minimum as well as maximum joint rail-water rates ».

7. That section 15 (4) be amended so as to permit the establishment of through routes regardless of the « short-hauling » of any carrier.

8. That Congress further consider the situation of steam railroads under the Revenue Act of 1936, « for the reasons stated in our report for 1936 under the heading « Sinking Funds and Other Reserve Funds ».

9. That Congress legislate to cover completely the standard time zone field.

10. That section 20 of the Interstate Commerce Act be amended so as to require carriers and their owners to furnish « correct » as well as « specific » answers to questions upon which the commission may need information.

11. That the commission be exempt from the amended provisions of section 11 of the Federal Register Act.

12. That amendments be adopted as discussed in the 1935 report regarding exemption from I. C. C. jurisdiction of street, suburban and interurban electric railways.

Before getting into its review of the work of various I.C.C. bureaus the report makes its general observations on various matters, including railway rate increases. Comment in the latter connection follows upon an assertion that it is not to be taken as indicating any attitude in regard to the pending Ex Parte 123 case on which the commission has « an entirely open mind ».

Continuing, the report says that it will be generally agreed « not only that the railroads are constitutionally entitled

to an opportunity to earn a reasonable return on the fair value of their carrier properties, but also that it is imperative that they be able, in general, to earn such a return under what may be called normal conditions, if the system of private ownership of railroads is to function at all satisfactorily. It will also be agreed, although perhaps not so generally, that if a reasonable return cannot be earned in times of depression, the railroads should have a somewhat corresponding opportunity to earn more than a reasonable return in times of prosperity, so that the average may be fair ».

Rate boost no ever-ready remedy.

Yet, the commission goes on to complain, many jump from these propositions to the conclusion that « to make them effective all that is needed is for the railroads to propose and for us to approve rates, fares and charges on an adequate level ». And although this idea « appears to have found root and to flourish in the minds of many owners of railroad securities and others who are not so well informed », those who are « most closely in touch with the practical aspects of the situation, like the active traffic officers of the railroads and of the large shipping industries, know that the problem is not so easily solved ».

There follows discussions of competition in transportation, of the railroads' failure to hold their pre-1929 share of the total traffic, and of the post-1929 decline in the average revenue per net ton-mile — all leading to the observation that « No competitive industry can work out its salvation through a price-increasing policy alone, and the railroad industry is now, to a very considerable extent in that class ». While it is conceded that price increases are « at times necessary », it is nevertheless asserted that « the vital thing in competition is to increase the salability of the product and reduce its cost ». Also, it is pointed out that aside from the competition of other agencies, a high level of transportation charges prompts large shippers « to reduce in every practical way the amount of transportation used ». And the commission has had « plenty of evidence that this has actually happened on a very considerable scale, through the establishment of branch factories and warehouses, economies in the use of fuel and other raw materials, and the like ».

Sees « manifest need for caution ».

The commission conceives it to be no part of its duty to interfere with management in the fixing of rates when the question is only one of what is wise or unwise in the exercise of sound business judgment; but it feels that conditions are now such that « there is a manifest need for caution in the increase of railroad rates, fares and charges, quite apart from the question of whether the increases will be lawful or unlawful ». Also, it believes that « in the exercise of sound business judgment there is like manifest need for doing everything that can be done to make railroad service attractive to the public and to improve earnings through reduction of operating costs ».

In the latter connection it is conceded that the railroads have done much in recent years; but it « is not clear that they have done all that can be done ». Here are recalled previous comments by the commission on co-ordination of facilities and the activities of the Federal Coordinator of Transportation — « an important reason » for permitting the expiration of which office was the creation of the Association of American Railroads, « with the object among others of bringing about these economies through voluntary action ». Yet, the report adds, « if there have been impor-

tant accomplishments by the association along these lines, they have not been brought to our attention ».

In its discussion of the traffic and earnings of transport agencies the report reviews briefly the trend of railway traffic and earnings, pointing out how freight ton-miles for the 12 months ended with July, 1937, were 9.1 % under the 1923-25 average and 17.7 % under the 1929 total. Also, if 1936 railway tonnage is compared with what it would have been had it advanced since 1923 in the same proportion as the nation's economic productivity, « a deficiency of nearly 200 million tons in railway freight traffic appears, to be accounted for partly by diversion to other transport agencies, partly by other economic changes, and to some extent possibly by imperfections in the various indices of production or in the statistics of commodities carried ».

The freight revenue shows a trend somewhat similar to that for ton-miles, « but less favorable because the course of the ton-mile revenue has been generally downward from 1923 to the present time ». The latter is said to have come about in the main from decreases in rates « partly by our orders but in large part by the voluntary action of the carriers ». Passenger traffic has incresed « with industrial recovery and a low fare policy » — for the first seven months of 1937 the number of passenger miles was 15.9 % above that for the corresponding 1936 period. The total passenger revenue has also increased since 1933, « notwithstanding the drastic reductions in fares ». In 1936 compared with 1935, the net revenue of the freight service increased and the net revenue of the passenger service decreased, « indicating a greater advance in revenues than in expenses in both services ».

No opinion on wage levels.

In referring to recent advances in wages, the commission expressed no opinion « as to the adequacy or inadequacy of the earnings of the employees », but it did observe that « such wage increases can presumably be met only in part by economizing in the use of labor ». Railway financial returns of recent months are given as reflecting some of the effects of wage increases as well as the recession in business. Next come the discussions of railway taxes, employment and net income after fixed charges. A table of 1936 gross and net earnings of carriers, other than motor carriers, reporting to the commission, shows that at the time « the upturn in operating revenues for steam railways was greater than for any other class in the table ». Figures for motor carriers were absent because these have not yet been required to file annual earnings reports. The commission, however, calls attention to the fact that motor vehicle registrations were higher in 1936 than in 1935 by 7.1 % for passenger cars and 10.3 % for trucks; that monthly consumption of gasoline was 10.3 % greater in the first six months of 1937 than in the corresponding 1936 period.

After a brief account of the commission's Fiftieth Anniversary celebration which was held April 1, 1937, the report proceeds through the above-mentioned discussion of railroad rate increases to brief comment on obsolete railroad plant, the effect on non-competitive rates of rate wars in the transport field, recent class readjustments rate, and drought and flood relief rates. In the latter connection mention is made of recent legislation relieving carriers of charges of discrimination growing out of their action in according such relief rates only to distressed areas and persons as defined in orders of the commission.

Free travel warrants study.

Information received in the first two of the quarterly reports on passes and

free tickets, which large steam railroads were required to file in 1937, makes it obvious to the commission that « the amount of money involved in free railway travel is sufficient to warrant careful study by the managements ». The returns show that the reporting railroads during the first six months of 1937 carried free 3 204 968 persons other than employees of the reporting carrier. This service involved a total of 601 million passenger-miles with an estimated value of $ 11 333 734. The latter is 5.36 % of the actual passenger revenue of the six months period, and for certain individual railroads the value of free transportation reported was « over 10 % » of the actual revenue from their revenue passengers. « It is not to be inferred », says the report, « that by discontinuing free passes to persons other than their own employees railways as a whole could secure additional revenue equal to that given above... However, the net effect on individual railways would vary since some are now giving more free service to other railways than they receive ».

Further special attention is given in the report to the co-operation of Federal and State commissions, the pending freight forwarding investigation, abandoned mileage and investigations by the commission. During the year ended October 31, 1937, the commission granted 116 abandonment applications involving 265.98 miles of main line and 723.075 miles of branch line of Class I roads, and 558.315 miles of short lines. Of the latter 335.8 miles constituted the entire lines of the applicants. Among the listing of its 1936-37 reports on investigations, the commission devotes considerable space to that on the New York, New Haven & Hartford, which became the vehicle for its recommendation that non-carrier railroad subsidiaries be brought within its jurisdiction. In discussing the latter among its legislative recommendations, the commission goes on to comment briefly on holding companies in connection with which it refrained from making any suggestions pending recommendations which may come from the Senate interstate commerce committee's investigation of railroad finances.

Discussing its recommendation for legislation requiring carriers and their owners to furnish « correct » as well as « specific » information, the commission argues for the proposed amendment by citing such instances as the Pittsburgh & West Virginia's alleged failure to disclose that it was controlled by the Penn-road Corporation and the « Special Deposit » device used by former Van Sweringen lines to record investments in unnamed properties.

Wheeler still using I. C. C. employees.

Senator Wheeler's investigation of railway finances still engages the services of a number of Bureau of Accounts accountants, the maximum number thus employed during the 12 months beginning November 1, 1936, being 56, and the average during the period being 37. The Wheeler probe, the commission feels, has shown « the need for equipping the commission with broader powers in investigation of financial practices and also with a larger staff of accountants ». Inadequate appropriations for the work of the Bureau of Accounts has forced the transfer of field men to the depreciation section, thus « further reducing a field force already undermanned ». During the year « much progress has been made in revising the accounting regulations for steam railroads, sleeping car companies, express companies and carriers by water ».

Actual issue of $ 664 460 635 of securities was authorized during the period under review, a large part of which was to meet maturities or to exchange for existing obligations. Thus the additional capitalization resulting from the

various authorizations amounts to $ 441 602 882, plus 809 450 shares of common stock without par value. Equipment trust certificates account for $ 184 337 000 of this amount. The report notes that during the period under review the market for railroad securities was not as favorable as during the period covered by the previous report. Yet, it adds that « some railroads which had securities that could be called for redemption were able to refinance and to market their securities, bearing lower rates of interest on advantageous terms to them ». Since the previous report the commission approved Reconstruction Finance Corporation Loans aggregating $ 20 098 805. Also, five additional proceedings for reorganization under section 77 of the bankruptcy act were launched; seventeen plans of reorganization have been filed since the 1935-36 report.

A total of 679 hearings were conducted in the period covered by the report on matters passing through the Bureau of Formal Cases. These involved 110 503 pages of testimony and compared with 725 hearings and 136 780 pages of testimony during the previous year. The Bureau of Inquiry conducted more than 200 investigations while the Bureau of Law gave its attention to 64 court cases involving commission orders or requirements. Activities of the Bureau of Locomotive Inspection and the Bureau of Safety, high-lighted in the report, will be covered in later reviews of their separate reports.

Bureau of motor carriers.

The commission devotes about 18 pages to the Bureau of Motor Carriers, opening with a discussion of the general progress of administration. The Bureau's « most pressing task » continues to be that of disposing of « grandfather-clause » applications; and the commission hopes shortly to announce « other procedures which will further lessen the delay which has occurred in disposing of these applications ». A total of 78 819 « grandfather-clause » applications were filed on or before February 12, 1936. Only 1 088 certificates or permits had been issued at the time of the report's preparation, although orders had been drawn granting certificates or permits in 4 480 cases and denying them in 2 878 others; 51 265 applications await field reports.

Discussions are included of the Bureau's work in connection with motor carrier tariffs; motor carrier safety regulations in which connection the cooperation of state authorities is acknowledged; and of leading motor carrier decisions of the year. Also, requirements for insurance or other security for the protection of the public. During the year under review the Bureau's Section of Traffic received 63 516 tariff publications of common carriers of passengers and property, and 2 053 schedules and 16 202 copies of written contracts and memoranda of oral contracts of contract carriers of property. Of the total 3 896 were rejected or returned as not being in accord with provisions of the act.

The Bureau's Section of Law and Enforcement has continued its policy of making every effort to secure voluntary compliance with the act, but it did secure 38 convictions in criminal cases and the entry of appropriate decrees in 14 civil cases. During the seven months ended October 31, 1937, the Section of Safety received motor carrier accident reports involving 3 878 injuries and 483 fatalities.

Motor bureau funds still inadequate.

Development of the Bureau's field organization has been retarded by lack of funds, the commission having received thus far only $ 2 450 000 for motor carrier work in the fiscal year 1938, it

requested an appropriation of $ 3 259 530. The field organization now has a total of 53 offices and six more are planned. One of the principal functions of the field staff at present « is to explain to carriers and shippers the provisions of the law and of our rules and regulations issued thereunder ». During the year the field staff addressed 350 groups, totaling approximately 15 000 persons; held 156 180 interviews with individuals; and wrote 166 260 letters. Many other activities « in which the field staff should engage » have been postponed « because of the primary necessity of determining the rights of grandfather applicants ».

Summing up this section the commission says that experience with motor carrier regulation « has demonstrated that our estimates of what was needed for the proper administration of the act were conservative ». It goes on to cite adverse effects on enforcement of reduced appropriations, and states that it will be unable, unless it receives additional funds, to undertake the work of investigating the need for safety regulations for private carriers.

Railroads did « excellent work » on grain movement.

The report's discussion of the Bureau of Service Activities pays tribute to the railroads and the Association of American Railroads « for the excellent work done in assembling cars in advance of the harvest, and in getting them back to loading points when released at terminals ». Reference is made to the suspension of the average per diem plan for which « no reason was given »; but « it is probable that the urgent necessity for the return of box cars to home lines for grain and other seasonal loading was a controlling one ». Between November 1, 1936, and October 31, 1937, surplus freight cars increased from 115 255 to 123 065 or 6.8 %. In the same period freight car ownership declined from 1 762 039 to 1 732 147; new units added totaled 67 783. The number of tank cars of fusion-welded construction approved for test purposes is now 367, and additional applications are under consideration.

The Bureau of statistics prepared the 50-year review of I. C. C. activities which is now being printed. An « important and growing » phase of this Bureau's work is the analysis of testimony and exhibits relating to the cost of transportation service. « This increased demand for cost data », the report says, « is in large part the result of keener competition between the railways and other means of transport ». Rate adjustments passing through the Bureau of Traffic were protested in 428 instances during the year; suspensions were granted in 163 cases and refused in 180 while 85 were otherwise disposed of. The year produced 544 fourth-section orders, 55 of which were denial orders. The Bureau of Valuation was engaged during the year « principally in meeting requests for valuation exhibits and analyses to be filed in proceedings for the reorganization of railroads... and in the valuation of pipe lines ». Because of a further reduction in its appropriation this Bureau's staff was reduced from 243 employees on November 1, 1936, to 212 on November 1, 1937.

During the first session of the present Congress, from January 5, 1937, to August 21, 1937, the commission's legislative committee submitted to Congress 85 reports on bills or resolutions; also, members of the committee testified on several occasions before Congressional committees.

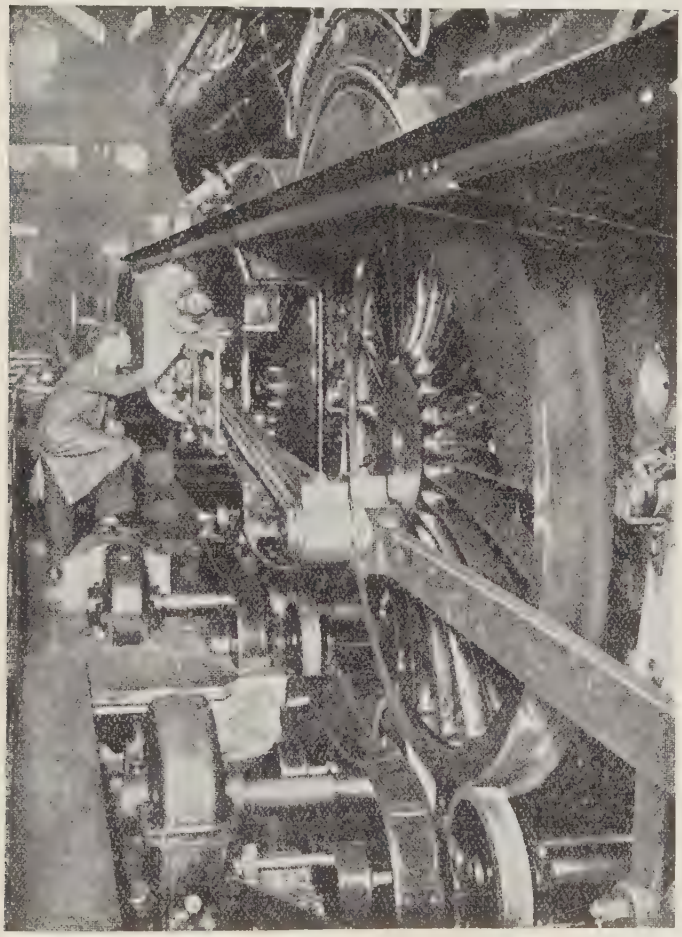
MISCELLANEOUS INFORMATION.

[621.134.2 & 621.9]

1. — An interesting locomotive valve-setting machine

(designed and built at Swindon Works, Great Western Railway).

(The Railway Gazette.)

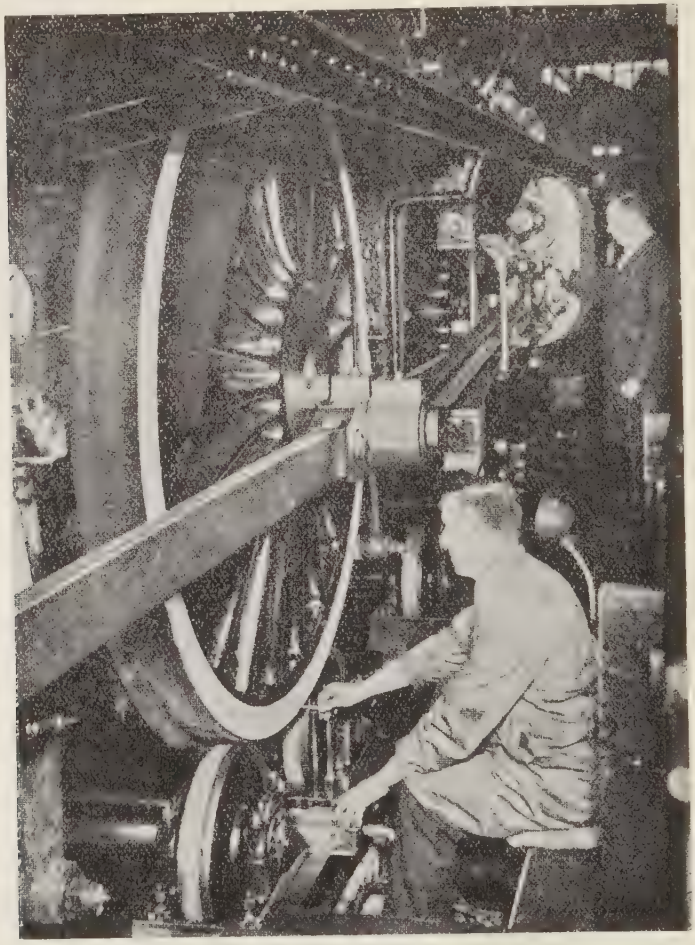


Machine controlled by operator near cylinder.
Portable push-button giving forward or reverse motion is in operator's left hand.

When setting the valves of a locomotive in the erecting shop, the usual method of moving the engine is that known as « pinching », a bar being used as a lever between the tread of the wheel and the rail; and although this may appear somewhat crude, it is still the

method most commonly resorted to in locomotive shops.

During a recent visit to the Swindon works of the Great Western Railway, we were greatly interested in the working of a machine which has been built and installed there, by which the accurate setting of locomotive valves is conveniently and expeditiously carried out. The machine, which is illustrated herewith, is accommodated in a special pit, and is driven by a 15-H.P. electric motor through reduction gear, suitable shafting, and worm gearing. The illustration shows a « King » class express engine on the machine. Two pairs of locomotive wheels are driven by rollers, and two pairs of idlers are provided for supporting the other wheels of a six or eight coupled engine. One pair of driving



Main control position. Trammel in operator's right hand from a fixed pillar registers dead centres from centre pops on rim of tyre. Press-button control box is within easy reach of his left hand.

rollers is fixed, and the remainder can be adjusted along the bed to suit the various axle centres.

The control is by press buttons — « forward », « reverse », « stop », and « inching ». These are situated near the right-hand driving wheel, so that the operator can mark and take the dead centres. A remote flexible push-button control is also provided at the right and left hand at the front of the engine, to enable the operator to control the machine when marking and taking the end of the piston stroke for dead centres and chart readings. The speed of rotation of a locomotive wheel of 5 ft. 8 in. diam., is one revolution in 68 seconds.

We had an opportunity whilst in the « A » erecting shop at Swindon, where the machine is located, of trying the controls, and found that the wheels of the locomotive could be moved in either direction by infinitesimal or larger degrees with the greatest ease and convenience. Our thanks are due to Mr. C. B. Collett, O. B. E., Chief Mechanical Engineer of the Great Western Railway, for facilities in connection with the inspection and photographing of the machine.

[621. 392 (.42) & 625. 143.4 (.42)]

2. — Automatic butt welding of rails,

London Passenger Transport Board.

(The Railway Gazette.)

In connection with the programme of the London Passenger Transport Board to provide quieter travel on the tube railways, a new portable plant for the end-to-end welding of rails into long lengths has just been acquired. Two bogie wagons comprise this plant, one being a mobile power station with a 400-h.p. diesel generating set, the other containing the welding set. The welding plant, which is automatic in operation, consists of an A-1 patent automatic flash welder of 300 kVA. rating, arranged with pneumatically operated clamps and upsetting mechanism. The standard 60-ft. lengths of running rail or conductor rail are lifted from the rail stacks on to a temporary conveyor, from which they are fed to the welding plant. The two lengths to be welded are placed in the jaws of the machines, aligned, and then clamped by heavy pressure exerted by the pneumatically-operated mechanism. The welding operation is then begun by push-button and no further attention is required by the operator until the weld is automatically completed, when the clamp jaws are released and the rails carried by the conveyor to rail wagons at the opposite end of the welding vehicle. The operation of the push-button to begin the welding switches on a heavy current, and starts a slow reciprocating movement of the right-hand welding head of the machine, which is mounted on slide bars and carries one of the rails, causing the two rail ends to be brought together and separated. The resultant arcing of the heavy current, as well as the resistance of the steel, heats the rail ends. The reciprocating action is maintained for about two minutes until the condition of the metal at the extremely is so altered as to permit of automatic change-over to flash welding, which is maintained for a further period of about ten seconds. This makes the rails sufficiently plastic for welding, the temperature being then about 3 500° C. and the maximum current flowing 45 000 amps. Again, entirely automatically, the current is switched off and the rail ends are forced together with a pressure of 27 tons to consolidate the joint. The complete welding operation thus comprises pre-heating, flashing, and upsetting, the whole cycle of operations being automatically controlled and completed in less than three minutes by simply operating a push-button.

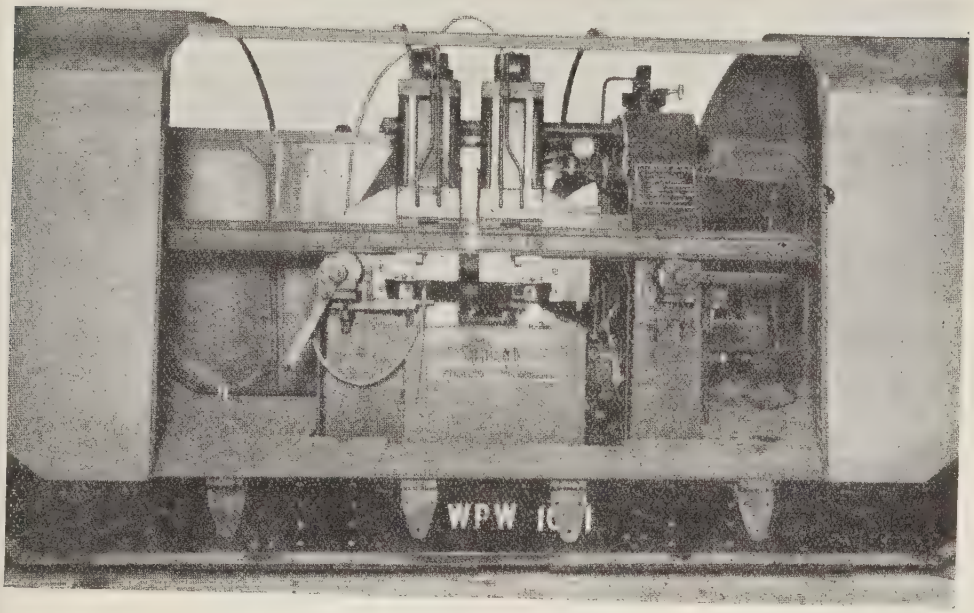
The welded rails are now taken by the

Portable 400 B.H.P. diesel-electric generating set and air-compressor set for flash-butt welding plant.

conveyor to the chipping plant, where the weld extrusion is removed, and then on to an electric post-heater or normaliser. The welded rails are placed in the jaws of this heater and securely clamped by pneumatically-operated mechanism. The current is switched on and when the required temperature of 850° C. is reached the heating current is automatically cut off, thereby ensuring that all joints receive the same post-heating treatment. The rails are then passed along the conveyor for final dressing and grinding of the welded joints, after which they are transferred to specially constructed rail wagons for transit, or stacked for future use. The complete operation of welding, post-heating, and dressing the joints, including handling, is done in about ten minutes so that the plant maintains an output of six welded joints an hour on 95-lb. R. B. S. running rails. The operation takes rather longer with flat-bottom conductor rails weighing up to 150 lb. per yd. There is, of course, no limit to the length of rail which can be built up by this means, but as the rails for the purposes of the London Passenger Transport Board have to be welded in a depot and then taken by train to the site it has been decided to form them into maximum lengths of 300 ft. For use in the open track it is proposed to have running rails made up into lengths of 240 ft., and conductor rails in lengths of 120 ft.

Comprehensive tests have been made on rails welded by this A-1 plant, and the test figures obtained on loads far in excess of running conditions have been highly satisfactory, indicating indeed that on the normal loading conditions the strength of the joint is up to the standard of unwelded rails.

The contract for the mobile welding plant was executed to the requirements of the London Passenger Transport Board by the A-1 Electric Welding Appliances Company, of Victoria Street, London, S. W. 1, and the welding machine and electric post-heater manufactured at the company's works at Inverness. The plant is designed for the automatic welding of running rails up to 12 sq. in. sectional area, and conductor rails up to 15 sq. in. area. The



A-1 flash-butt welder mounted on wagon, showing 95-lb. running rails in position for welding.

clamps of the 300-kVA. automatic post-heater for normalising the joint after welding are arranged for pneumatic operation and automatic regulation, and the temperature is regulated by optical pyrometer control (manufactured by Radiovisor Parent Limited). The normaliser is mounted on wheels and can be brought into the required post-heating position at will.

For the supply of alternating current to the welder and post-heater, a 400-B.H.P. diesel engine and 300-kVA. alternator, manufactured by the English Electric Co. Ltd., are included in the equipment. The two special well wagons on which the welder and the generating plant are respectively mounted, were designed and manufactured by London Transport. On the generator wagon the air compressing unit for the operation of the post-heater and the grinding and chipping auxiliary tools are situated. A self-contained motor-driven air compressor unit for the welding process is mounted on the welding wagon. Both compressor units were manufactured by Broom & Wade Limited.

The mobility of the plant is of particular interest. The two wagons have been designed to negotiate open track and tube tunnels at a maximum speed of 30 m.p.h. Welding sites are being arranged at various points on the system and the complete units can be transported as required.

NEW BOOKS AND PUBLICATIONS.

[313 : 656]

KELLERER (Dr. Hans). — **Mathematik und Verkehr** (Mathematics and communications). — A pamphlet (4 3/4 × 7 inches) of 48 pages, with 48 figures. — 1938; Publisher : B. G. Teubner, Leipzig and Berlin.

In this little book the author has endeavoured to give some examples of the application of statistical methods. The original data are taken from reports on very different methods of communication. It is explained how the basic figures collected, which have little meaning in themselves, can be combined mathematically or graphically to furnish new elements making interesting comparisons possible.

The graphical method is the one most frequently used. No special knowledge is required, neither about communications nor statistics. Mathematical analysis is only called upon once, the method of the least squares being used for drawing down the curve representing the movement of telegraphic traffic from 1925 to 1934.

The book is an excellent means of arousing interest in statistical research. Communications of all kinds have developed so rapidly in our times that they have become an almost inexhaustible source of study.

Those who wish to pursue their researches will find ample matter in the statistical sources indicated in the appendix, as well as in two larger volumes, one of which by the same author (Verkehrsstatistik, Berlin 1936 — see *Bulletin of the Railway Congress*, January 1937, p. 305).

E. M.

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[656]

TREVISANI (Renato), Professor of the University of Trieste. — **La Concorrenza fra la strada e la rotaia nella politica economica dei trasporti** (Competition between road and rail in the transport economic policy). — One volume (6 × 9 inches) of 352 pages. — 1937, Milan; Ulrico Hoepli, publisher. (Price : 28 lire.)

The question of road competition is one of the primary preoccupations of railway administrations. It has been the subject of reports and discussions at several Congresses and it is often evoked when examining operating methods or principles. No wonder, therefore, that it should have attracted the attention of well informed and experienced writers who wish to put it in its proper sphere and define the methods most likely to solve it.

In the *Bulletin of the Railway Congress* for October 1937 we gave a brief analysis of the valuable study by Sr. F. Tajani and described by the way the measures taken in Italy. These consist in taxing motor transport and devoting the proceeds of the tax to meeting the railway deficit. Other details about the position in Italy and the remedies suggested or already tried were given in this review. The October 1937 *Bulletin* gives the principles of the law of the 20th June, 1935 — XIII, regulating goods transport in motor vehicles, and further information will be found in the very detailed notes by Mr. Ibl on this same subject of regulation.

In Professor Trevisani's book the whole subject is considered and investigated under all its aspects. As the title

indicates, the question is not merely one of competition of road transport against the railway, but of the whole general transport policy.

The starting point is the examination of the position of the railways as regards monopoly. To show how thorough the author has wished to make his studies, we will give some of the sub-headings of Chapter I : Transport considered as a public service; public operation and private operation; the institution of concessions; economic basis of the rates; real cost and average cost; differential rates; political and social obligations weighing upon the railways.

In the second chapter, the author describes the birth and development of motor competition, its effects combined with those of the economic crisis, and the reactions of the railways. He analyses the action taken by the latter as regards the rates and reports various systems of regulating the relations between the two methods of transport.

The third chapter reviews the attempts at co-ordination made in other countries, and the legislative measures adopted.

Chapter IV describes the land transport policy adopted in Italy. Here the reader will find a complete report of the solutions adopted in that country together with the reasons therefor. Though it is difficult to sum up in a few words such a complicated situation and the whole of so many measures, it seems possible to state that the leading ideas are based on the application or reintroduction of a system of monopoly together with distribution of the traffic.

The leading principles are fully discussed again in Chapter V : Co-ordination in the corporate system.

The last part of the book is a valuable documentary appendix, wherein the author has collected the text of the principal laws on rail and road co-ordination in European countries.

E. M.

The Universal Directory of Railway Officials and Railway Year Book, 1938-39. — London : The Directory Publishing Co. Ltd., 33, Tothill Street, Westminster, S. W. 1. — One volume (8 1/2 in. × 5 1/2 in.) of 604 pages. (Price : 20 sh. net.)

The Universal Directory of Railway Officials originally issued in 1895, reaches with the present volume its forty-fourth year of publication.

By fixing the date of issue in the middle of the year it has been found possible to include most of the operating and other figures for the year 1937, and thus give the volume the maximum currency with the latest annual information.

In the new edition, opportunity has been taken to include a chronological list of outstanding events in railway history.

The increasing development in the electrification of main-line steam railways has again necessitated revision of the tabulated information on this subject, and the table of the world's total railway mileage has been completely overhauled and amended from official replies to a questionnaire.

There are three indexes, namely, (1) an index to countries; (2) a general index, including all reference to railways and statistical and other information; and (3) a personal index of railway officials.

The lists of railway officials have been brought up to date, together with brief description of the chief railway governmental and other authorities exercising control over railways.

[625. 215]

SALKADE (K. N.). — **Points and crossings, with explanatory notes and formulæ, for the 5 ft.-6 in. gauge.** Third edition, revised and enlarged. — A volume (7 1/2 × 4 3/4 inches) of 374 pages with 97 illustrations. — 1937, Bombay (India), The Popular Book Depot, Grand Road, No. 7.

Modern track is submitted to static and dynamic stresses which have increased continually during the last years in consequence of the growing axle loads and increasing speeds.

Operation under high speeds demands a carefully laid and systematically maintained track.

Mr. Salkade is the author of three books, the whole of which will prove very useful for the Indian and other permanent way staff.

These works are :

1. A volume devoted to the study of the permanent way properly speaking, track laying and maintenance, and upkeep of bridges and other structures (« Permanent way text book for all gauges ») ;

2. A volume dealing with « Points and crossings for the metre and narrow gauges » ;

3. A special volume treating points and crossings for the 5 ft.-6 in. gauge (see heading).

Track laying demands sound reasoning and reflection.

It is no longer possible to rely on earlier « rule of thumb » ideas in this matter.

Elementary geometry and trigonometry are indispensable to those who wish te tackle these questions.

Mr. Salkade has succeeded in imparting the desirable notions in a condensed, though very clear, form.

These enable him to deal methodically with all standard cases, from the ordinary turnout to the complicated appliances used in large modern stations (cross-over roads, diamond crossings, single or double-slip points, scissor cross-over roads, etc.).

The Author concludes his work by examining certain special cases, as the result of his own experience on the Indian Railways.

To sum up, Mr. Salkade's three books make up an excellent documentation available for the permanent way staff, as well as for use in the schools educating such staff.

J. D.

[385. (09 (. 497.1)]

MARTIN (Dr. Friedrich). — **Eisenbahngeographie Jugoslawiens** (Geography of the Jugoslavian Railways). — One volume (6 × 8 1/4 inches) of 90 pages, with 4 inset maps. — 1937, Verlag Wissenschaftlicher Werke Konrad Triltsch, Würzburg.

The layout of the Jugoslavian railway system is the result above all of the political changes undergone by the countries that today form this kingdom. In the first part of his book, the author gives the history of the building of the various railway lines.

He divides it into four periods :

The first, which ends in 1878, saw the connecting up of Vienna and Budapest with Trieste and Fiume respectively by lines which are only transit lines as far as Jugoslavia is concerned.

From 1878 to 1900, and from 1900 to 1918, the construction of lines of military interest is witnessed, as well as intense competition between districts and countries with conflicting aspirations.

After the great War, the authorities endeavoured to organise and complete the system to conform it to the new frontiers, to the orography, and to the new general directions of the traffic currents.

The author describes the characteristics of the railway system in the second part of his book, and analyses the traffic.

In the third part, he studies the diverse influences of railway communications on the development of populations, and economic geography. This leads him to discuss proposed new lines intended to supply connections that are in great request.

The density of the system varies considerably from one part of the country to another. In certain provinces it is more than 5 miles per 100 square miles; in others it is less than 2.5 miles. It is very curious to see how, apart from the factors mentioned above, it depends on the general configuration of the mountain ranges and the plains separating them, as well as upon the distribution of the population. To what extent do the railways meet the requirements of industry and commerce ? This question and others are dealt with by the author in the fourth and last part of his study.

The improvement and adaptation of the railways to the new conditions has been one of the leading preoccupations of the Government of the country, owing to their great importance from the point of view of peace and its security.

In reading this book, one is struck by the close connection between the life of nations and the development of the means of transport, together with the remarkable part played by the railway in the political and social events this country was the scene of, for nearly a century.

E. M.